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THE STIRLING ENGINE FOR RAILWAY LOCOMOTIVE APPLICATION

REPORT NO. T P 2930E

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THE STIRLING ENGINE FOR RAILWAY LOCOMOTIVE APPLICATION

by

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for

Transport Canada
Transportation Development Centre
Montreal
Quebec
Contract No. 4270

February 1981

Report No. T.P. 2930E

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Ottawa, Canada K1A 0N5

REPORT DOCUMENTATION FORM

1. Transport Canada Report No.	2. TDC Project No.	3. Recipient's Catalogue No.
T.P. 2930	4270	

4. Title and Subtitle	5. Report Date
The Stirling Engine for Railway Locomotive Application	
	6. Performing Organization Report No. None Given
7. Author(s)	8. Transport Canada File No.
G. Walker	D500-642-1
9. Performing Organization Name and Address	10. DSS File No.
Whitegaites Engineering Ltd. 4023 Vardell Road N.W. Calgary, Alberta T3A 0C3	11. DSS or Transport Canada Contract No. 4270
12. Sponsoring Agency Name and Address	13. Type of Report and Period Covered
Transportation Development Centre 1000 Sherbrooke St. W. P.O. Box 549 Montreal, Que. H3A 2R3	14. Sponsoring Agency Code
15. Supplementary Notes	16. TDC Project Officer R.H.F. Dillon

17. Abstract

Future shortages and high costs of fuel oil are expected to result in the use of coal for railroad locomotives. A brief study has been made of the Stirling engine for this application. A Stirling engine is a heat engine receiving heat from an external source, converting a fraction of it to useful work and rejecting the remainder as waste heat. It was invented in the 19th century and was widely used until displaced by the internal combustion engine and electric motor at the turn of the century.

Redevelopment of the Stirling engine began about 40 years ago. It is widely used for cryogenic cooling and advanced prototypes up to 150 kW have been developed for automotive use.

Conceptual engineering studies of large (1000 horsepower) Stirling engines have been made in the United States. A comparison of these with contemporary locomotive diesel engines shows that Stirling engines of equivalent power and efficiency could fit the same installation envelope as direct replacement power units in existing locomotives.

Fluidized bed coal combustors would be located beneath the locomotive frame where the oil fuel tanks are normally located.

Novel proposals are made for Ringbom-Stirling engines using air as the working fluid and water as the lubricant with power boost at maximum power output by high pressure steam injection. Such engines could be made by converting existing locomotive diesel engines using the same parts and components. Enormous savings would result from the gradual conversion of existing locomotive fleets to coal-burning operation rather than outright replacement as would be required for steam locomotives.

18. Key Words	19. Distribution Statement
Stirling, Engine, Railway, Locomotive, Air Engine, Coal, Fluidized Bed, Heat Pipe	A limited number of reports are available.

20. Security Classification (of this report)	21. Security Classification (of this page)	22. No. of Pages	23. Price
Unclassified	Unclassified		

TDC CDT 79-005

REPORT DOCUMENTATION FORM

1. Transport Canada Report No.	2. TDC Project No.	3. Recipient's Catalogue No.
T.P. 2930	4270	

4. Title and Subtitle	5. Report Date
The Stirling Engine for Railway Locomotive application	6. Performing Organization Report No.

7. Author(s)	8. Transport Canada File No
G. Walker,	D1465-345-1

9. Performing Organization Name and Address	10. DSS File No.
Whitegaites Engineering Ltd. 4023 Vardell Road N.W. Calgary, Alberta T3A 0C3	11. DSS or Transport Canada Contract No 4270

12. Sponsoring Agency Name and Address	13. Type of Report and Period Covered 1981
Centre de développement des Transports 1000 ouest, rue Sherbrooke - 25e étage Montréal, Québec H3A 2R3	Final, décembre 1980-février 14. Sponsoring Agency Code

15. Supplementary Notes	16. TDC Project Officer
	R.H.F. Dillon

17. Abstract Les pénuries prévisibles et les coûts élevés du carburant provoqueront peut-être l'emploi du charbon dans les locomotives de chemin de fer. A cette fin, une étude sur le moteur STIRLING a été entreprise. Il s'agit d'un moteur thermique recevant sa chaleur d'une source extérieure, convertissant une partie de la chaleur en force thermique et rejetant le reste en perte de chaleur. Ce moteur a été inventé au 19e siècle et a été en usage jusqu'à la fin du siècle lorsqu'il a été remplacé par les moteurs à combustion interne et électrique. Ce moteur a de nouveau été utilisé depuis une quarantaine d'années. Il est largement utilisé pour la réfrigération et des prototypes élaborés jusqu'à 150 KW ont été créés pour les véhicules à moteur. Aux Etats-Unis, des études conceptuelles d'ingénirie du gros moteurs Stirling (1000 HP) ont été entreprises. Une comparaison avec les moteurs diesel contemporains indique que ces moteurs Stirling, à puissance et à capacité égales, pourraient s'adapter facilement pour remplacer les moteurs thermiques dans les locomotives présentement en usage. Les éléments de combustion du charbon sur un lit fluidisé seraient placés sur le chassis des engins de locomotive diesel où les réservoirs de mazout sont habituellement localisés. La nouveauté de l'idée consiste à utiliser un moteur Ringbom-Stirling qui utiliserait l'air comme fluide de travail et l'eau comme lubrifiant. La puissance de sortie maximale serait obtenue par suralimentation en injectant de la vapeur à pression élevée. L'avantage est que les locomotives diesel d'utilisation courante pourraient être converties aux moteurs à cycle de Stirling, tout en gardant le même emballage. Ainsi, il serait possible de réaliser de très fortes économies en dotant graduellement le parc actuel de tels moteurs, plutot que de remplacer carrément ce parc par des locomotives à vapeur.

18. Key Words	19. Distribution Statement
Cycle de Stirling. Moteur thermique. Chemin de fer, Locomotive. Moteur à vapeur Charbon. Lit fluidisé.	Le CDT dispose d'un nombre limité d'exemplaire.

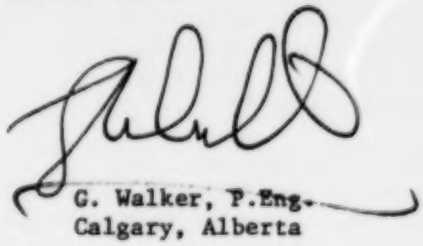
20. Security Classification (of this report)	21. Security Classification (of this page)	22. No. of Pages	23. Price
Non classifée	Non classifée		

ACKNOWLEDGEMENTS

Helpful information and advice about the Series 251 diesel engine was received from Bombardier Inc./MLW, Montreal, Quebec.

FOREWORD

This limited study of the potential for application of coal-burning Stirling engines to railroad locomotives in the context of Canadian transportation was carried out under contract with Transport Canada, Transportation Development Centre, Montreal, Contract No. 4270. The project officer for Transport Canada was Mr. Richard Dillon. Tne study was performed during December 1980/February 1981.



G. Walker, P.Eng.
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SUMMARY

Future shortages and high costs of fuel oil are expected to result in the use of coal for railroad locomotives. A brief study has been made of the Stirling engine for this application.

A Stirling engine is a heat engine receiving heat from an external source, converting a fraction of it to useful work and rejecting the remainder as waste heat. It was invented in the 19th Century and was widely used until displaced by the internal combustion engine and electric motor at the turn of the century.

Redevelopment of the Stirling engine began at Philips Laboratories, Eindhoven, some 40 years ago. Now the Stirling engine is widely used for cryogenic cooling and advanced prototypes up to 150 kW have been developed for automotive use by the Swedish licensees, United Stirling.

Detailed engineering studies of large (1000 horsepower) Stirling engines have been made in the United States. A comparison of these proposed engines, modified for locomotive use with contemporary locomotive diesel engines shows that Stirling engines of equivalent power and efficiency could fit in the same installation envelope as direct replacement power units in existing locomotives.

Fluidized bed coal combustors would be located under the locomotive frame between the trucks where the oil fuel tanks are normally located. Sufficient space would remain for a small buffer coal store fed by screw auger from a coal tender towed by the locomotive.

Novel proposals are made for Ringbom-Stirling engines using air as the working fluid and water as the lubricant with power boost at maximum power output by high pressure steam injection. Such engines could be made by converting existing locomotive diesel engines using the same parts and components. Enormous savings would result from the gradual conversion of existing locomotive fleets to coal-burning operation rather than outright replacement as would be required for steam locomotives. Recommendations are made for further engineering designs of the Ringbom-Stirling engine including the development of a 20 horsepower proof-of-principle demonstration engine converted from an industrial two-cylinder diesel engine. Estimated cost and time of the programme is $250,000 per year for 3 years.

Part II of the report contains a brief appraisal of gas-fired Stirling engines for locomotive propulsion including some aspects of liquid energy gas handling, storage and the hazards attendant therein.

LE SOMMAIRE

Insuffisance de futur et grands coûtes d'huile combustible sont attendront résulter d'usage du charbon pour locomotives du chemin de fer. Une étude bref a fis du moteur Stirling pour cette application.

Un moteur Stirling est un moteur de chaleur que recoit chaleur de source externe, convertit une petite portion à travail utilement et rejecte l'autres dans chaleur gaspillement. Il était inventé dans le 19th siècle et il a grande utilisation jusqu'à displacement par le moteur combustion interne et le moteur électrique au commencement du cet siècle.

Redéveloppement du moteur Stirling a commencé au Laboratorie Philips, Eindhoven depuis 40 ans. Maintenant, le moteur Stirling est utiliser grande pour frigorifique cryogenique et des originals advancés jusqu'à 150 kW ont developé pour l'usage automotive des patentés suédois, United Stirling.

Etudés de l'ingénieur en details des grands (1000 h.p.) moteurs Stirling ont fis dans l'états unis. Une comparaison des ces moteurs proposés, modifié pour l'usage locomotive avec locomotive contemporaine moteur diesel suggére que les moteurs Stirling de puissance équivalent et efficacité ont pu ajuster dans la même enveloppe d'installation comme puissance unité direct remplacement dans locmotives existés.

Charbon combusteur de fluidisé véhicule sont siteué sous la cadre de locomotive entre les trocs où le réservoir d'huile combustible sont situent normalment. Espace suffisant a resté pour une petite volume provision de charbon nourrit de tarière vis du tender de charbon remorqué de la locomotive.

Des propositions originals sont fait pour des moteurs Ringbom-Stirling et employent d'air pour le fluide de travilé et de l'eau pour la lubrifiant avec augmentation de puissance de injection de vapeur haut poussée à rendement maxime puissance. Lês memes moteurs pu sont faire largement de conversion de moteurs diesels existés que employe les mêmes pièces et composants. Economies colossal résulteront des conversions graduels des flottes de locomotives existés à fonctionnements charbon brûler plutôt que remplacement complet demandé pour locomotives vapeurs. Recommandations sont faire pour plus avant-projet de technique de l'ingénieur du moteur Ringbom-Stirling qui comprend le développement du 20 h.p. démonstration moteur épreuve principe convertit d'un diesel moteur industriel deux-cylindre.

Coût estimatif et temps du programme est $250,000 par an pour trois ans.

RÉSUMÉ

Les pénuries prévisibles de carburant et les coûts de plus en plus prohibitifs provoqueront peut-être l'emploi du charbon pour la propulsion des locomotives de chemins de fer.

Dans cette éventualité une étude succinte a été effectuée du moteur à combustion externe, ou moteur à cycle de Stirling. Il s'agit d'un moteur thermique recevant sa chaleur d'une source extérieure, convertissant une partie de la chaleur en force thermique et rejetant le reste en perte de chaleur. Ce moteur dont l'invention remonte au 19e siècle a été très répandu jusqu'à l'avènement du moteur à combustion interne et du moteur électrique, au début du 20e siècle.

Le moteur Stirling a été développé, il y a environ quarante ans par les laboratoires Philips, à Eindhoven, dans les Pays-Bas. De nos jours, il est largement utilisé pour la réfrigération cryogénique et des prototypes élaborés d'une puissance de 150 KW ont été créés pour les véhicules à moteur par United Stirling, concessionnaire suédois.

Aux États-Unis, des études détaillées ont été effectuées, portant sur des moteurs Stirling capables de développer (1000HP). Une comparaison de ces moteurs avec les moteurs diesel contemporains indique que les moteurs Stirling, à puissance et à capacité égales, pourraient s'adapter facilement pour remplacer les moteurs thermiques dans les locomotives présentement en usage, tout en occupant le même volume d'espace.

Les éléments de combustion du charbon sur un lit fluidisé seraient placés sous le châssis de la locomotive, là où les réservoirs de mazout sont habituellement installés. Il resterait assez de place pour un petit magasin à charbon, servant de tampon et alimenté par vis sans fin à partir du tender.

La nouveauté de l'idée consiste à utiliser un moteur Ringbom-Stirling qui utiliserait l'air comme élément de puissance et d'eau comme lubrifiant,

avec une puissance de pouvoir à son maximum par l'injection de vapeur à haute pression. De tels moteurs pourraient être fabriqués en convertissant les moteurs actuels de locomotives Stirling utilisant les mêmes pièces et les mêmes composants. Ainsi, il résulterait de très fortes économies en transformant graduellement l'équipement de locomotives vers la combustion en charbon plutôt que de le remplacer totalement par des locomotives à vapeur. Le rapport recommande la poursuite de l'étude du moteur Ringbom-Stirling et la mise au point, à titre démonstratif, d'un moteur de 20HP obtenu par la conversion d'un moteur diesel courant à deux cylindres. Le coût prévu est de 750 000$ répartis également sur une période de 3 ans.

La partie 11 du rapport fait une appréciation succinte du moteur à cycle de Stirling utilisant un combustible gazeux et appliqué à la propulsion de locomotives et, aborde divers aspects connexes de la question: manutention, stockage et danger du combustible liquéfié.

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1. INTRODUCTION

A minor study has been made, under contract with Transport Canada, of the potential for application of Stirling engines to transportation in the Canadian context. This was interpreted to refer more specifically to the potential of coal-burning Stirling engines as the power unit for railway locomotives in place of oil-fuelled diesel engines. There was never any suggestion this substitution might occur in the near future. It is an option for the longer term when conventional oil supplies become limited in supply and expensive, while coal remains abundant and relatively cheap. The rapid rise in oil prices of the 1970's has already generated an appreciable economic advantage for coal (about a quarter of the cost per unit of heat value). Further gains in oil costs are expected sometime in the future to result in compelling economic imperative as well as strategic considerations regarding security of supply.

Synthetic oil can be produced from coal to allow continued use of diesel engines. Another possibility is to burn pulverized coal, probably as an oil slurry, in a diesel engine. Yet again, external combustion engines using coal directly with minimal processing and suitable for railway locomotives include the reciprocating steam engine and the Stirling engine. Coal-fired steam engines were, of course, the overwhelmingly dominant form of locomotive propulsion unit in North America until displaced, virtually entirely, by diesel engines in the 1950's. The steam engine remains the major locomotive propulsion unit in parts of the world with plentiful supplies of coal and wood, but little or no indigenous oil, South Africa, India, South America, Eastern Europe.

The Stirling engine has never been used for locomotive propulsion, so far as is known. Furthermore, only experimental prototype Stirling engines have been constructed having power outputs even approaching those customary with railway locomotives. Nevertheless, the future of coal-burning Stirling engines appears bright indeed both in new locomotives and as a more or less direct replacement to the oil-burning diesel engine in the many thousands of road and yard locomotives operating today. It appears possible to fit a coal-burning Stirling engine of equivalent power and efficiency into the same space as a conventional oil engine. The Stirling engines would use many of the same engine components as the diesel engine they replace and would operate silently using air as the working fluid and water as the engine lubricant. The engines could be installed in existing diesel locomotives with a fluidized bed coal combustor contained centrally under the locomotive frame between the trucks (bogies) where the fuel tanks are normally found on diesel locomotives. A small 'buffer' coal supply could also be accommodated therein but road units would likely haul a coal tender.

A shortage of diesel oil in the future appears inevitable and the limited supply available will be expensive. Moreover, all known processes for coal-to-oil conversion are expensive and require substantial 'refinery' type chemical processing facilities. Therefore, it is highly probable the direct use of coal as the fuel for locomotive propulsion will become a matter of pressing economy and the choice lies between steam and

Stirling traction engines or the use of coal in stationary engines for the production of power for electrified railway transport. This latter is preferred for relatively short distance, dense traffic corridors but the capital cost is prohibitive for long distance, intermittent traffic operation. The use of direct fuelled locomotives will remain for these lines.

It was in the above context of railway futures that the present study was undertaken of Stirling engines for locomotive propulsion.

2. STIRLING ENGINES

The Stirling engine is a form of heat engine that receives heat at high temperature from an external source, converts some of the heat to mechanical work and rejects the balance as waste heat at a low temperature.

Figure 1 is a cross-section of an idealized Stirling engine. It consists of a cylinder containing a displacer and a piston. The space above the displacer, called the expansion space, is heated to a high

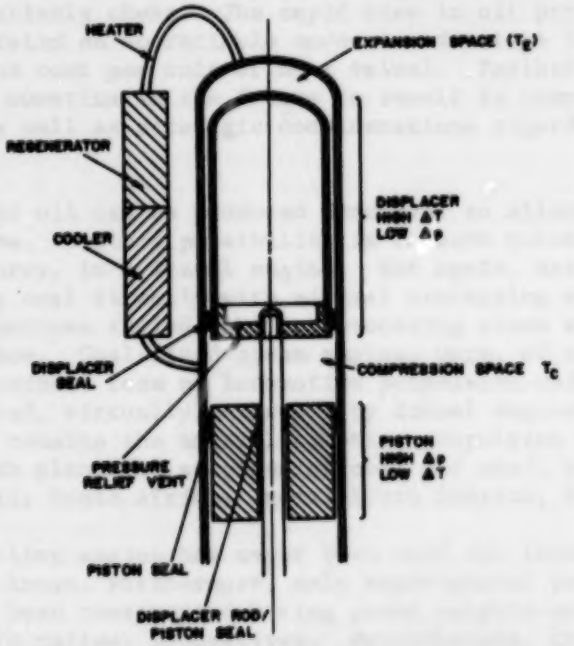


Figure 1 Elements of a Stirling Engine.

temperature, $T_{(max)}$. The space between the displacer and the piston is called the compression space and is maintained at a low temperature close to ambient, $T_{(min)}$. The volume of the two spaces varies cyclically with the motion of the piston and displacer as shown in Figure 2. The volume variations of the expansion space lead those of the compression space by about a quarter of a cycle, i.e. 90° of crank rotation.

The two spaces are coupled by a duct so the pressure is substantially the same in the two spaces and as the volumes of the spaces vary some of the working fluid passes from one space to the other. The connecting duct contains three heat exchangers termed the heater, the regenerator, the cooler. The heater is a recuperative heat exchanger with slots or fine tubes for the passage of the working fluid. It is the primary unit for the transfer of energy at high temperature *to* the working fluid from the energy source, a combustion unit, radioisotope, solar energy or thermal store.

The cooler is similarly equipped with slots or fine tubes for passage

of the working fluid and also the cooling medium, usually water at ambient temperature.

In operation, the working fluid is concentrated in the hot expansion space and the pressure is high when the piston is descending (the working or expansion stroke). Conversely the displacer is at the top of its stroke with the working fluid concentrated in the compression space so the pressure is lower when the piston is ascending, (the compression stroke). The cyclic pressure variation is approximately sinusoidal and so phased with the piston motion to develop the kidney shaped pressure-volume or work diagrams shown in Figure 3.

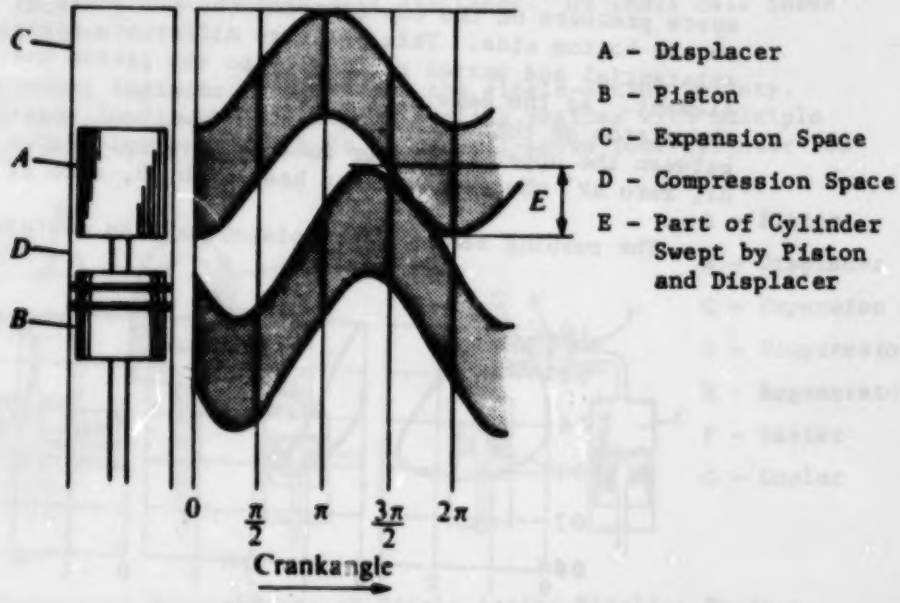


Figure 2 Volume Variations in a Stirling Engine.

There is a net transfer of energy at high temperature through the heater to the engine. Part of the energy is converted to mechanical work at the piston for useful application and the remainder is rejected from the engine through the cooler at approximately ambient temperature.

The regenerator is another heat exchanger interposed between the heater and the cooler. It consists of a porous matrix of finely divided material, metal strips, wires, sintered powder, etc. It may be thought of as a thermodynamic sponge alternately accepting and rejecting heat from the working fluid.

The displacer and piston both reciprocate in the cylinder and appear, superficially, to have much in common. There are significant differences to justify the use of separate names. The open duct connecting the expansion and compression spaces assures that the pressure above and below the displacer is the same except for minor pressure losses in the regenerator and other heat exchangers. The displacer is therefore essentially

a no-work element simply displacing fluid from the compressor to expansion spaces or vice-versa. The displacer seal is subject only to a small pressure difference and furthermore some leakage across the seal is tolerable although for proper operation of the engine the main flow of working fluid must always be directed through the heat exchangers.

Although the displacer has no significant pressure difference it experiences the full temperature difference between T_{max} at the top side and T_{min} at the bottom. To minimize thermal conduction the displacer is therefore a long thin-wall element of relatively low structural strength and light in weight.

The piston sustains the full pressure difference between the working space pressure on the top face and the crankcase or bounce space pressure on the bottom side. This pressure difference across the piston is very substantial and varies cyclically so the piston must be a strong structural element. At the same time there is no great temperature difference across the piston so that it can be designed without regard to thermal conduction between the upper and lower faces. In simple terms a displacer has 'high ΔT, zero ΔP' whereas a piston has 'high ΔP, zero ΔT'.

The rubbing seals of the piston have to contain the working fluid in

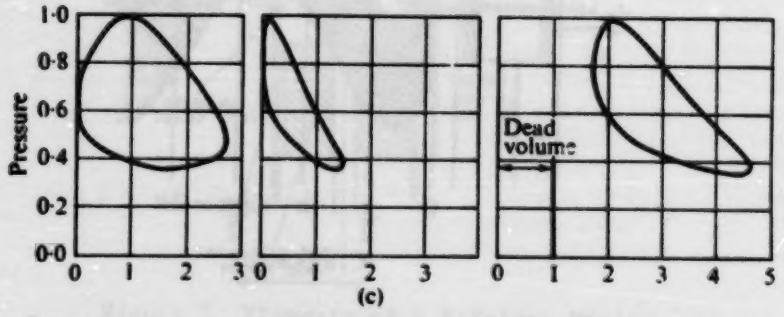


Figure 3 Work Diagrams for the Stirling Engine.

the working space. They serve another, more demanding, function; to prevent egress of lubricating oil into the working space where it eventually degrades and blocks the fine interstices of the regenerator. There are two piston seals shown in Figure 1, one for the piston and cylinder, another for the piston and displacer rod. These piston seals are the Achilles heel of modern high performance Stirling engines and will exert substantial influence in our subsequent selection of systems for application to railroad locomotives.

2.1 Mechanical Arrangements

The above discussion has served to introduce the essential elements of a Stirling engine as two spaces at different temperature levels, T_{max} and T_{min}, whose volumes can be varied out of phase so the hot expansion

space leads the cold compression space. The two spaces are coupled through a regenerator and two other heat exchangers, a heater and a cooler.

These elements can be arranged in endless variety but can be generally classified in two groups as:

a) single-acting,
b) double-acting.

Single-acting machines comprise two reciprocating elements, two pistons or a piston and displacer for each Stirling cycle system. Most versions can be further categorized into one of the three different types shown in Figure 4, where the piston and displacer are contained in the same cylinder, the tandem arrangement, or in separate cylinders. Another family has a second piston instead of a displacer. We shall call these 'two-piston machines'.

Small Stirling engines are always of the single-acting variety. Large engines can also be single-acting Stirling engines with multiple units arranged on a common crankshaft. One well-known four-cylinder 200

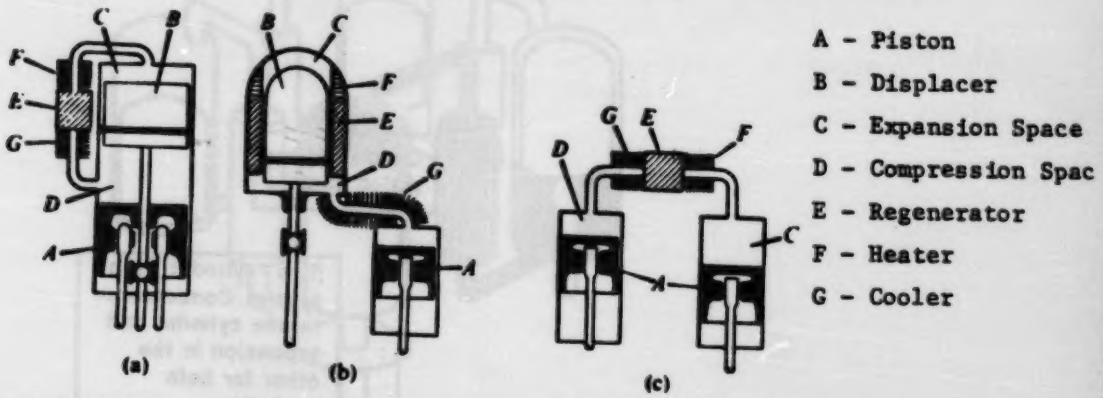


Figure 4 Mechanical Arrangements of Single-Acting Stirling Engines.

horsepower Stirling engine is shown in Figure 5. This was made in the 1960's by the Philips Company in Holland as a bus engine. It is basically four single-acting Stirling engines of the tandem piston/displacer variety arranged on a common crankcase. The complicated drive system is called a 'rhombic drive'. This engine proved too expensive to manufacture commercially.

The other major group of Stirling arrangements are the double-acting types represented in Figure 6. Here multiple cylinders are connected successively so the expansion space of one cylinder, above the reciprocating element, is coupled to the compression space below the reciprocating element, of the adjacent cylinder. Each Stirling system requires only one reciprocating element, a piston-displacer, so that major economies are possible compared with ensembles of multiple single-acting machines. Anything from 3 to 7 cylinders may be used. One favourite combination called the 'square four' coupling is shown in Figure

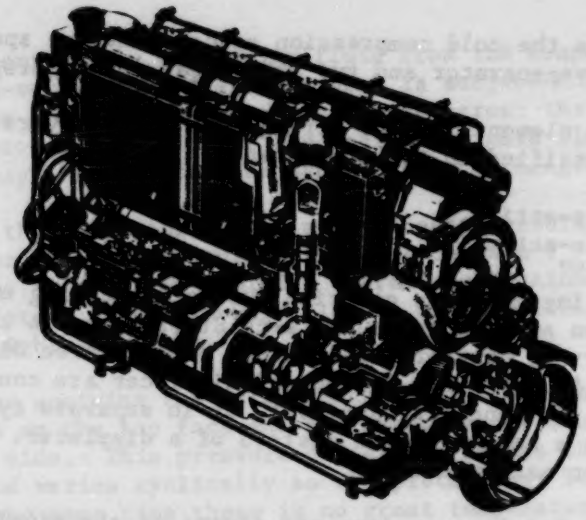


Figure 5 Four-Cylinder, In-Line, Single-Acting Stirling Engine with Rhombic Drive, (courtesy Philips Co.).

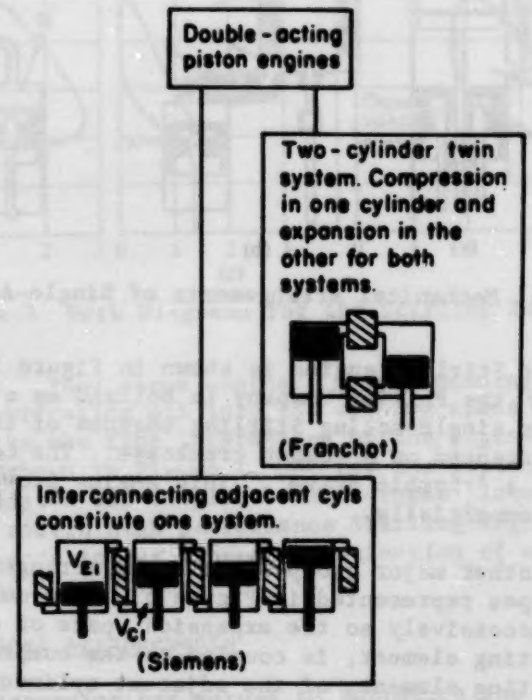

Double - acting piston engines
Two - cylinder twin system. Compression in one cylinder and expansion in the other for both systems.
(Franchot)
Interconnecting adjacent cyls constitute one system.
V_{E_1}
V_{C_1}
(Siemens)

Figure 6 Double-Acting Stirling Engines.

7. Figure 8 shows a cross-section of an advanced prototype engine of 75 kW, developed by United Stirling, a Swedish licensee of the Philips company and said to be scheduled for production in 1982 with initial application said to be intended for mining applications. The same engine is used in the study of Stirling automotive engines conducted by MTI/ United Stirling/American Motors for the United States Department of Energy/NASA.

2.2 Free-Piston Engines

Both Figures 5 and 8 show the pistons coupled to some kind of drive mechanism. Such arrangements are generically called 'kinematic drives' and include the simple crank-slider (connecting rod/crank), rhombic drive, Scotch yoke and various other ingenious mechanisms.

Other forms of Stirling engines have no kinematic drive but simply

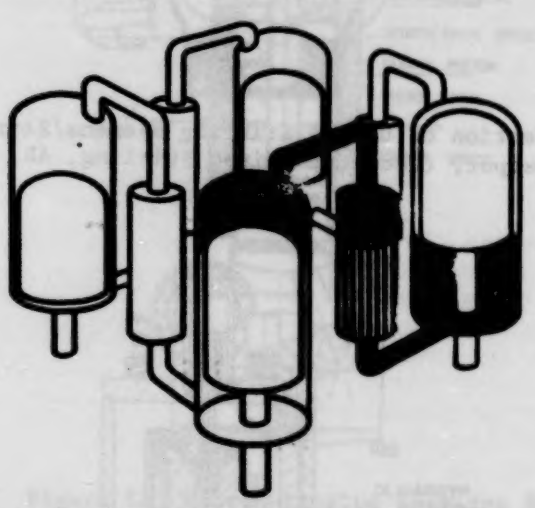


Figure 7 The 'Square-Four' Siemens Stirling Engine Arrangement.

operate by virtue of the balance of fluidic forces acting upon the reciprocating elements. These are called free-piston Stirling engines and were invented by Professor William Beale in the early 1960's. They have been progressively developed since then, principally by Sunpower Inc., of Athens, Ohio, a company founded by William Beale. There are many different forms of free-piston Stirling engines. Figure 9 is a cross-section of an early Sunpower demonstration engine in commercial production for most of the 1970's. It could be supplied driving a linear electric power generator, or water pump. Several free-piston Stirling engines are in limited production at Sunpower for gas-fired heat pumps, water pumps and electric power generation. The largest made to date has a power output of 3 kW.

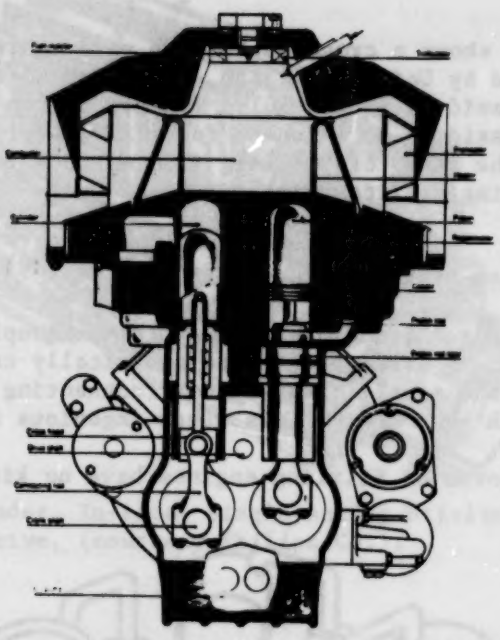


Figure 8 Cross-Section of United Stirling Siemens/Stirling Engine of 75 kW Output, (courtesy United Stirling, AB, Malmo).

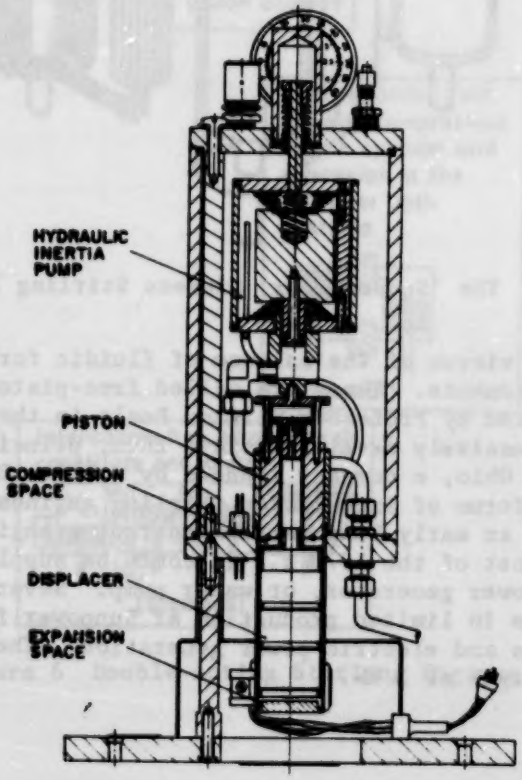


Figure 9 Cross-Section of Early Sunpower Free-Piston Stirling Engine, (courtesy Sunpower Inc., Athens, Ohio).

2.3 Hybrid Ringbom Engines

Another variant of this basic system has the piston coupled to a kinematic drive (a simple crank-connecting rod) and a free displacer as shown in Figure 10. This is called a hybrid or Ringbom engine after the inventor Ossian Ringbom, a Russian living in Finland who took out a U.S. patent on the engine in 1905. So far as is known the Ringbom engine was never commercially developed until it was reinvented in the 1960's.

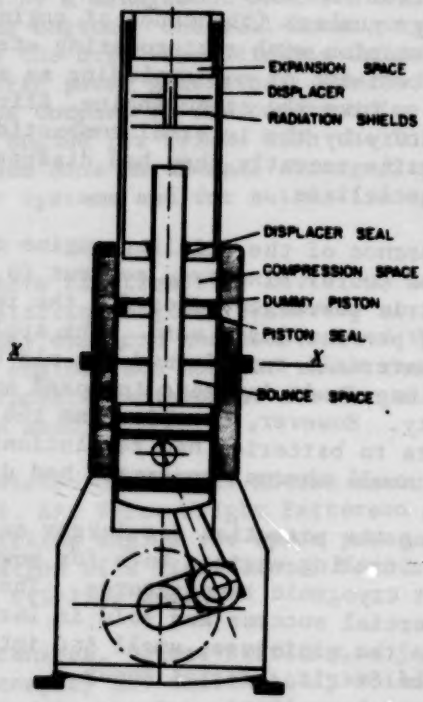


Figure 10 Hybrid-Ringbom Stirling Engine.

It is presently used widely in miniature form for the 'Split-Stirling' cryocoolers of infra-red night vision and heat seeking missile guidance systems (see Walker 1981).

More recently, a 1 kW Ringbom Stirling engine was developed by Sunpower in 1980 under contract to a major United States engine manufacturer.

The principal attraction of the Ringbom engine is the use of an uncomplicated kinematic drive connecting the piston to a rotating shaft. This permits use of presently available internal combustion engineering parts and techniques, particularly the use of rotary electromagnetic generators. A significant problem with purely free-piston Stirling engines is that linear electric generators are large heavy units which completely negate the advantage of the small compact system inherent in a free-piston arrangement.

A Ringbom engine permits the combination of much existing internal combustion parts and practices with the relative simplicity of free-piston Stirling engines. Recognition of the potential for conversion of existing diesel locomotive engines to Ringbom hybrid Stirling engines is perhaps the major contribution of this study.

2.4 History

Stirling engines formerly known as 'hot air engines' have a long history dating from the late 18th Century. They were developed and manufactured in large numbers (thousands of engines) throughout the 19th Century in competition with reciprocating steam engines. They had the attractive characteristic of not exploding as steam boilers of the day were wont to do. Like the steam engine, Stirling engines were displaced in the 20th Century by the internal combustion engine and the electric motor. Until quite recently they had disappeared from the purview of all except a few specialists.

The *renaissance* of the Stirling engine commenced over 40 years ago when the Philips Co. of Eindhoven set out to develop a small thermally activated electric power generator for the relatively formidable power requirements of pre-war radio sets. The application of modern design knowledge and materials transformed the old hot air engines and resulted in an order of magnitude increase in speed and two orders of magnitude in power density. However, by this time the invention of the transistor and improvements to batteries had revolutionized radio requirements and the need for a small thermal generator had disappeared.

Recognizing the potential for larger engines, Philips continued the development of Stirling engines both for power and for refrigeration, specifically at cryogenic temperatures. The Stirling cryocoolers were an immediate commercial success and sold in large numbers. Cryogenic applications in the miniature, small and intermediate range are the principal use of the Stirling engine today.

Subsequent development of the Stirling engine by Philips for power included a family of single-cylinder engines from 1 to 100 horsepower. They concentrated attention on engines of 40 to 50 horsepower per cylinder and the piston-displacer tandem arrangement with rhombic drive. For most of the 1960's they directed their effort to marine applications but at the request of a new licensee, United Stirling, converted the flat opposed piston four-cylinder Boxer marine engine to the four-cylinder in-line bus engine shown in Figure 6.

Later in concert with the other licensees Philips switched to the development of double-acting engines. Philips favoured a swash-plate kinematic drive and throughout the 1970's cooperated with the Ford Motor Company in the development of a Stirling automotive engine. About 1975 funding for the development was provided by the United States Department of Energy but in 1978 Ford abandoned both the gas turbine and Stirling engine to focus their development effort on stratified charge spark ignition engines. Shortly thereafter (1980) Philips abandoned further research/development activity on Stirling engines; a tragic end to over 40 years

sustained development effort at the level of some 50 engineers and their support staff resulting in impressive technical achievement.

General Motors were the first licensee of the Philips Co. and worked vigorously on the Stirling engine from 1957 to 1970. The G.M. Technical Centre, Warren, Michigan, was the prime focus of research activity and following a decade of intense component and system development were in the final stages of a four-cylinder bus engine assembly when the programme was abandoned by a management decision unrelated to the technical merit of the Stirling engine. One well-known product of the Technical Centre programme was the U.S. Army Stirling Ground Power Unit, a silent 10 horsepower, electric power generator. The Technical Centre also co-operated with Philips Company in the construction of a four-cylinder 360 horsepower Stirling engine for trials with the United States Navy. Much other pioneer work was done on thermal storage/Stirling engine combinations for underwater power systems and for automotive applications of Stirling engines.

The General Motors Electromotive Division, La Grange, Illinois, were also interested in Stirling engines primarily as the propulsion unit for coastal vessels. They designed and constructed various eight-cylinder 800 horsepower Stirling engines but none of these reached an advanced prototype stage before the corporate decision to abandon all development of Stirling engines was made.

The Allison Division of General Motors executed a research/development contract for the U.S. Air Force Wright Patterson Air Force Base for a 5 kW solar powered Stirling engine for space power. This engine was notable as the first application of a liquid metal heat transfer loop coupling a thermal source to a Stirling engine.

The Swedish licensees, United Stirling, a joint venture of the Swedish government defence company FFV and the ship-building company Kockums, of Malmo, was established in the late 1960's and continued development of Stirling engines for automotive application throughout the 1970's. With the departure of Philips from the field, United Stirling have become the prime centre of continuing Stirling engine activity. They have concentrated primarily on 40 kW and 75 kW double-acting engines with a 150 kW engine made by combining two of the 75 kW systems on a common crankcase. These engines are said to be scheduled for production for 1982.

The financial collapse of the Kockums corporate conglomerate in 1979 and its subsequent take-over by the Swedish government means that United Stirling is now wholly government owned and future directions are unknown.

In 1977 United Stirling teamed with the U.S. companies, Mechanical Technology Inc. of Latham, New York and American Motors, to become the 'second source' for the DOE/NASA automotive Stirling programme. This effort continued after the collapse of the Philips/Ford programme and is now the only large-scale government supported research programme on Stirling engines in North America.

Another licensee, formed jointly by the West German diesel builders MAN-MWM in 1967, worked on Stirling engines for vehicular and marine applications. In the early to mid-1970's they published extensively about their work but latterly have concealed all evidence of their activity. It is said that they are presently concentrating on 'large' (800 horsepower) multiple-cylinder Vee and in-line double-acting engines, but no details of these engines are known.

Other notable Stirling research programmes include the 20 years development of free-piston Stirling engines by William Beale of Ohio University and Sunpower Inc. of Athens, Ohio. Stirling engine artificial hearts have been developed over the past 15 years by teams at the Aerojet Energy Conversion Co., Sacramento, Ca. and the Joint Centre for Graduate Study, Richland, Washington. These programmes were funded by the National Institutes of Health, Washington, D.C.

Details of all the above and much other activity related to Stirling engines for power, university research, and minor programmes will be found in the recent major literary work on Stirling engines (Walker 1980).

A similar scale of development effort on Stirling and other cryocoolers in the past 25 years has been recently summarized (Walker 1981). An extensive treatment of the history and technology of free-piston Stirling engines has been prepared (Walker *et al* 1982).

Japanese involvement in the Stirling engine field is quite recent but has accelerated rapidly to become perhaps the major concentrated effort of the contemporary scene. A summary of Japanese programmes and interest in Stirling engines, particularly from the railway aspect, was commissioned for this study and is included in Appendix 1. The summary was prepared by Professor Yoshihiro Ishizaki of the University of Tokyo, Department of Nuclear Engineering.

A review of contemporary European developments was also commissioned and is included as Appendix 2. This was prepared by R.A. Billett of Radan Associates Ltd. of Bath, England.

3. LARGE STIRLING ENGINES

Comparatively little work has been done on large high power Stirling engines (arbitrarily defined here as a power output greater than 1 MW). No Stirling engines of this size have ever been constructed or even designed in detail. Much of the engineering evaluation of large engines that has been made has never been published.

The first significant contribution was the publication, by Rolf Meijer (1970), of the results obtained in a study, by Philips, for a 900

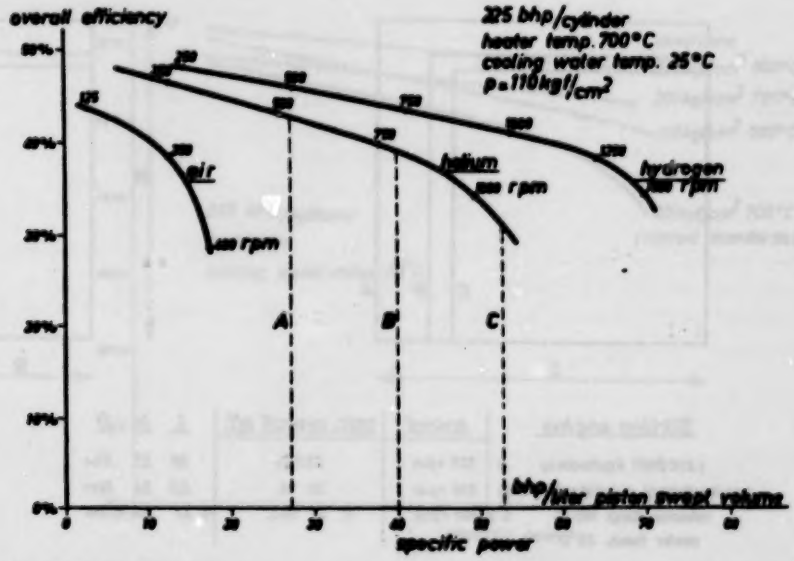


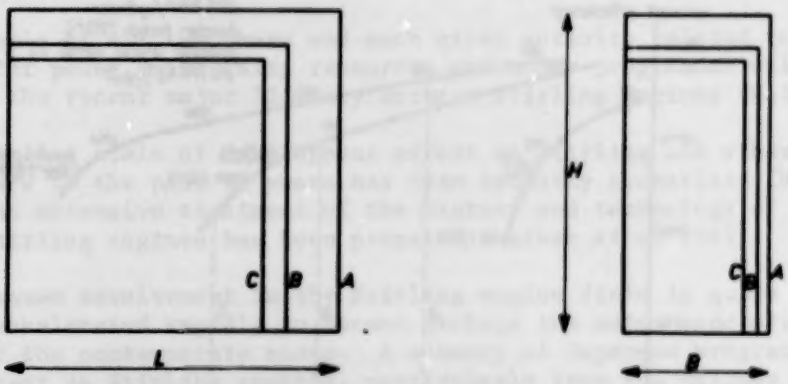
Figure 11 Calculated Performance of Stirling Engines with Different Working Fluids. Engines have the same output of 165 kW (225 brake horsepower) per cylinder and have been optimized for the maximum possible efficiency with heater and cooler temperatures of 700°C and 25°C and a mean pressure of 110 kg/cm. Working fluids are air, hydrogen and helium, (after Meijer 1970).

horsepower, four-cylinder Stirling engine. Some of Meijer's data is reproduced in Figures 11, 12 and 13. Figure 11 shows the maximum possible overall efficiency as a function of specific power (brake horsepower per litre of piston displacement volume) for a single-acting engine with rhombic drive and a power output of 225 brake horsepower per cylinder, heater temperature 700°C, cooling water temperature 25°C and a mean pressure of the working fluid of 110 kg/cm^2. Characteristic curves are given for hydrogen, helium and air as the working fluid. The corresponding engine speeds are given at intervals along the characteristics.

These results were used to determine comparative size envelopes shown in Figure 12 for a four-cylinder 660 kW (900 brake horsepower) stationary engine using helium as the working fluid at the three different operating states shown in Figure 11, corresponding to 525, 810 and 1260 revolutions per minute.

Figure 13 is an extension of the results given in Figure 11 for hydrogen as the working fluid and illustrates the gains in maximum possible efficiency resulting from increase in the heater temperature (from 700 to 800°C) and mean pressure (from 110 to 220 kg/cm^2).

Contemporary diesel locomotive engines are typically rated at 200



Stirling engine		speed	calc. overall eff.	L	H	B
(900 BHP, 4 cylinders)	A	525 rpm	43.5%	28	27	17m
helium P_m = 110 atm	B	810 rpm	39 %	23	24	18m
heater temp. 700°C	C	1260 rpm	31 %	21	23	896m
cooler temp. 25°C						

Figure 12 Comparative Size Envelopes for a 660 kW (900 Brake Horsepower) 4-Cylinder Engine with Helium as the Working Fluid at the Three Different Operating States Shown in Figure 11. Engines are rhombic drive machines, (after Meijer 1970).

horsepower per cylinder so Meijer's Stirling engine is comparable and later, size comparisons are made of 3000 horsepower Meijer-type engines and contemporary diesels.

The Electromotive Division of General Motors at La Grange, Illinois (the well-known diesel locomotive builders) in the mid 1960's investigated large Stirling engines of 800 to 1000 horsepower capacity. They constructed some experimental units including a one-half, four-cylinder bank of an 8-cylinder Vee engine of 800 horsepower engine. These studies and the results obtained were never published. Sketches of different engine concepts and a photograph of the one-half Vee engine were given by Hoagland *et al* (1978).

The referenced report by Hoagland was commissioned by the United States Department of Energy to evaluate the technology of Stirling engines for power generation in the 600 to 2000 horsepower range. It is the prime reference for the large Stirling engine field because the other principal author was Mr. Worth Percival. Percival, a man of exceptional ability directed the General Motors Stirling engine research effort throughout the 1960's and, following their withdrawal from the field, became a Stirling engine consultant primarily associated with the Swedish United Stirling development programme. In any directory of Stirling engine expertise Percival would rank among the top five. Therefore, because of his association with it, the Hoagland *et al* (1978) report deserves the closest study. It is a comparative survey of all aspects of large stationary power generation with Stirling engines utilizing coal-burning fluidized bed and stoker systems and indirect heat transport systems including

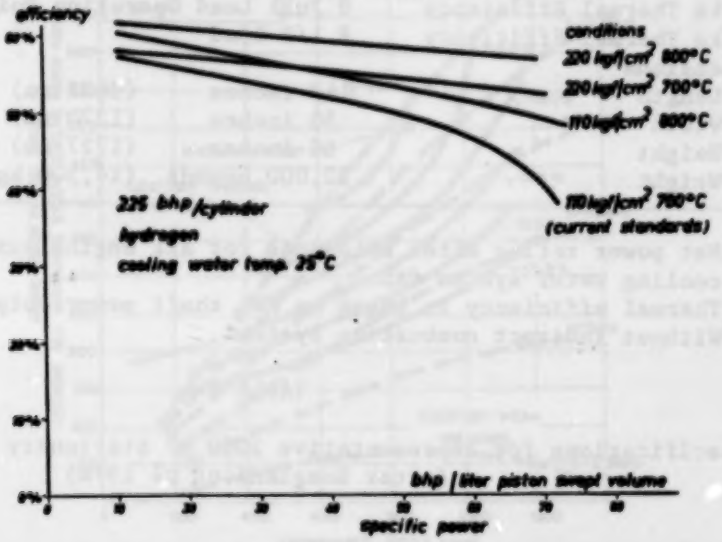


Figure 13 Effect of Maximum Cycle Pressure and Temperature on the Maximum Possible Efficiency Versus Power Density Characteristics for Engines with Hydrogen as the Working Fluid with a Brake Power Output of 165 kW (225 Brake Horsepower) per Cylinder and a Cooler Temperature of 25°C, (after Meijer 1970).

a sodium liquid/vapour heat pipe.

The summary and discussion of the technological risk areas of the Hoagland report are reproduced in Appendix 3 of this document. Table 1 also reproduced from the Hoagland report contains an outline specification of a 16-cylinder stationary power Stirling engine of 2000 horsepower output. This engine is a combination of four 4-cylinder Siemens 'square-four' Stirling engines in a twin crank, 8-cylinder in-line-per-crank arrangement, of same style as the United Stirling engine shown in Figure 8. The specification was obtained by scaling the established geometry and performance of the United Stirling P75, 75 kW engine. The scaling rules for

Power Rating	2000 bhp* - Continuous @ 850 rpm	
	2600 bhp - Intermittent @ 1130 rpm	
No. Cylinders	16 (four, 4-cylinder double-acting units)	
	125 bhp per cylinder (continuous)	
Cylinder Arrangement	Twin-crank - 8 cylinders in-line per crank	
Bore	7.18 inches (182 mm)	
Stroke	4.0 inches (102 mm)	
Heater Temperature	800°C (1470°F)	
Cooling Water Temperature	30°C (86°F)	
Displacement	162 in^3 (2.65 liter)/cylinder	
Working Gas	Helium	
Mean Operating Pressure	2030 psi (14 MPa)	
Combustion System	Indirect firing with sodium heat-pipe	
Fuel	Can be furnished with either liquid fuel or solid fuel combustion system	
Brake Thermal Efficiency**	@ Full Load Operating Point - 39.6%	
Brake Thermal Efficiency	@ 1/2 Load - 35.6%	
Dimensions***		
Length	143 inches	(3632 mm)
Width	50 inches	(1270 mm)
Height	68 inches	(1727 mm)
Weight	32,000 pounds	(14,500 kg)

* Net power rating after allowance for all engine auxiliaries except cooling water system fan.
** Thermal efficiency is based on net shaft power output ÷ fuel LHV.
*** Without indirect combustion system.

TABLE 1

Specifications for Representative 2000 HP Stationary Stirling Engine (after Hoagland *et al* 1978)

large Stirling engines given in the Hoagland report are reproduced here in Appendix 4. We shall use the same rules for scaling to 3000 horsepower. The estimated performance map for the basic 500 horsepower, four-cylinder module is given in Figure 14.

The Hoagland report was followed by the U.S. Department of Energy initiating three contracts for the design and development of Stirling engines for stationary power generation in the 500 to 3000 horsepower range. The programme was described by Holtz (1979) of the Argonne National Laboratory designated by DOE to be the programme manager. Phase 1 of the programme involved the preparation of state-of-the-art conceptual designs having the greatest potential for prototype testing for 1985. Advanced conceptual designs based on new technologies requiring significant research and development were also included in phase 1. The engine was to be integrated with a variety of energy sources, including fluidized bed-coal combustors and a suitable heat transport system.

Three parallel contracts for this Phase 1 design exercise were funded by the Department of Energy. The three groups involved were:

a) Advanced Mechanical Technology Inc., Newton, Mass., principal contractor in association with United Stirling, Malmo, Sweden, Ricardo Consulting Engineers, England and Thermacore Inc., Leola, Penn.

b) General Electric Co., Space Division, Philadelphia, Penn. in association with the General Electric Locomotive Division, General Electric Energy Technology, Schenectady, New York, Foster Wheeler Development

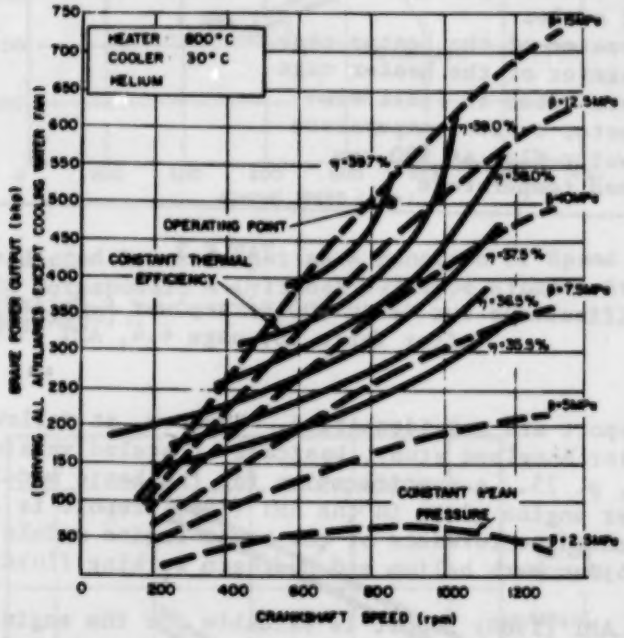


Figure 14 Estimated Performance Map for Representative Stationary Stirling Engine Basic 500 Horsepower Four-Cylinder Power Unit, (after Hoagland *et al* 1978).

Corp., Thermacore Inc., Leola, Penn. and, most significantly, Dr. Meijer of Philips.

c) Foster-Miller Association, Boston, Mass., principal contractor in association with Sunpower Inc., Athens, Ohio.

At the time of writing, final reports for these studies have not been published but two interim reports of exceptional interest are at hand, (Advanced Mechanical Technologies Inc. (1980) and General Electric (1970)).

The first of these references, the first quarterly report on the contract, is basically a development of the Hoagland report. It was prepared by the same organization although specific contributors to the AMT

Number of cylinders per module	4
Bore	200 mm
Stroke	110 mm
Swept volume per cylinder	3,455 cm^3
Number of heater tubes per cylinder	24
Outside diameter of heater tube	14.5 mm
Inside diameter of heater tube	10 mm
Heater tube length	850 mm
Number of regenerators per cylinder	4
Diameter of regenerator	120 mm
Height of regenerator	140 mm
Number of coolers per cylinder	4
Length of cooler	210 mm
Inner diameter of the heater cage	500 mm
Outer diameter of the heater cage	1,000 mm
Gap between tubes in first row	1.86 mm
Cooling water outlet temperature	50°C
Cooling water flow at 900 rpm	26 kg/s
Heater head temperature	720°C

TABLE 2

Specifications for Basic 500 Horsepower 4-Cylinder Engine Module
(after Table 10, page 4.4, AMT 1980)

(1980) report are not identified. However, it follows the same path as the earlier Hoagland study, basically a scaled version of the United Stirling, p. 75. A specification for the basic module of 4-cylinder 500 horsepower engine given in the AMT (1980) report is reproduced in Table 2. The estimated performance of the basic engine module is given in Figures 15 and 16 for both helium and hydrogen working fluids.

The AMT (1980) report is valuable for the engineering details it contains of the alternative heat transport systems including sodium heat pipe, forced convection gas and liquid, and the direct fired and other coal-burning heat sources.

The General Electric (1980) report is a substantial single volume subtitled task 1A, 'State-of-the-Art Conceptual Design'. It is a document of exceptional interest and value, highly relevant to this present study. For this reason Section 1, Summary and Overview and part of Section 2, State-of-the-Art Conceptual Design, are reproduced in Appendix 5.

The fundamental engine characteristics of the General Electric engine are given in Table 3. They favoured a Siemens double-acting Stirling engine of 125 horsepower per cylinder in a four-cylinder module of 500 horsepower. They systematically examined a number of different

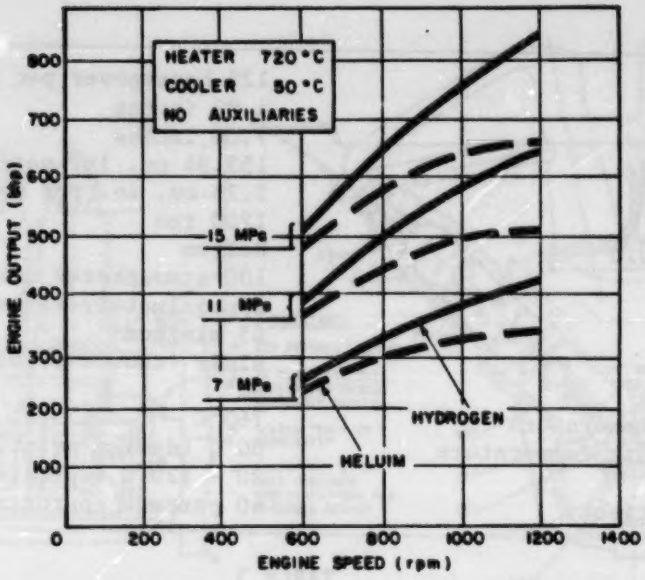


Figure 15 Predicted Power Output as a Function of Speed for a Basic 500 Horsepower, 4-Cylinder P75 Type Siemens Stirling Engine, (AMT 1980).

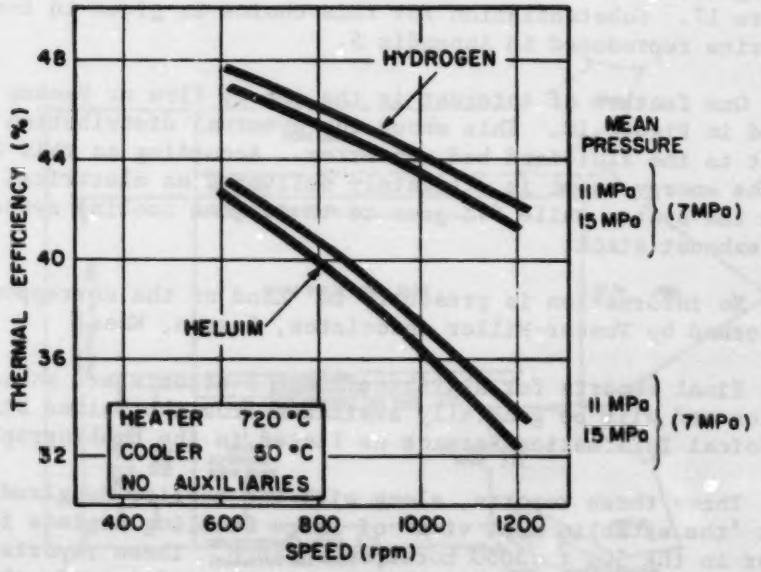


Figure 16 Predicted Overall Efficiency as a Function of Speed for Basic 500 Horsepower 4-Cylinder Type Siemens Stirling Engine, (AMT 1980).

Power	125 horsepower per cylinder
Stroke	4.00 inches
Bore	7.00 inches
Displacement	153.94 cu. in. per cylinder
	1.23 cu. in. per horsepower
Speed	1200 rpm
Working Fluid	Helium
Pressure	100 atmospheres mean, approximately 135 maximum, 65 minimum
Engine Type	Rinia (should be Siemens) *(Walker)*
Phase Angle	90°
Hot Side Temperature	760°C
Coolant Inlet Temperature	80°C (design point)
	20 - 120°C (operating range)
Shaft Efficiency	40 percent approximately

TABLE 3

Specifications of Basic Engine Parameters for Siemens-Stirling Vee Engine of 125 Horsepower per Cylinder Rating (after G.E. 1980)

configurations and selected the 90° Vee articulated arrangement shown in Figure 17. Substantiation for this choice is given in the summary and overview reproduced in Appendix 5.

One feature of interest is the energy flow or Sankey diagram reproduced in Figure 18. This shows the eventual distribution of the energy input to the fluidized bed converter. According to this diagram, 26.5% of the energy input is ultimately delivered as electrical power output from the system while 54% goes to the engine cooling system and 15% to the exhaust stack.

No information is presently to hand of the corresponding project performed by Foster-Miller Associates, Boston, Mass.

Final reports for all these Phase 1 efforts are scheduled for publication and will be generally available from the United States National Technical Information Service as listed in the bibliography.

These three reports, along with the earlier Hoagland report, represent 'the establishment view' of large Stirling engines for stationary power in the 500 to 3000 horsepower range. These reports are very substantial efforts by highly competent teams and justify the closest attention in any study of the Stirling engine for railway locomotive use.

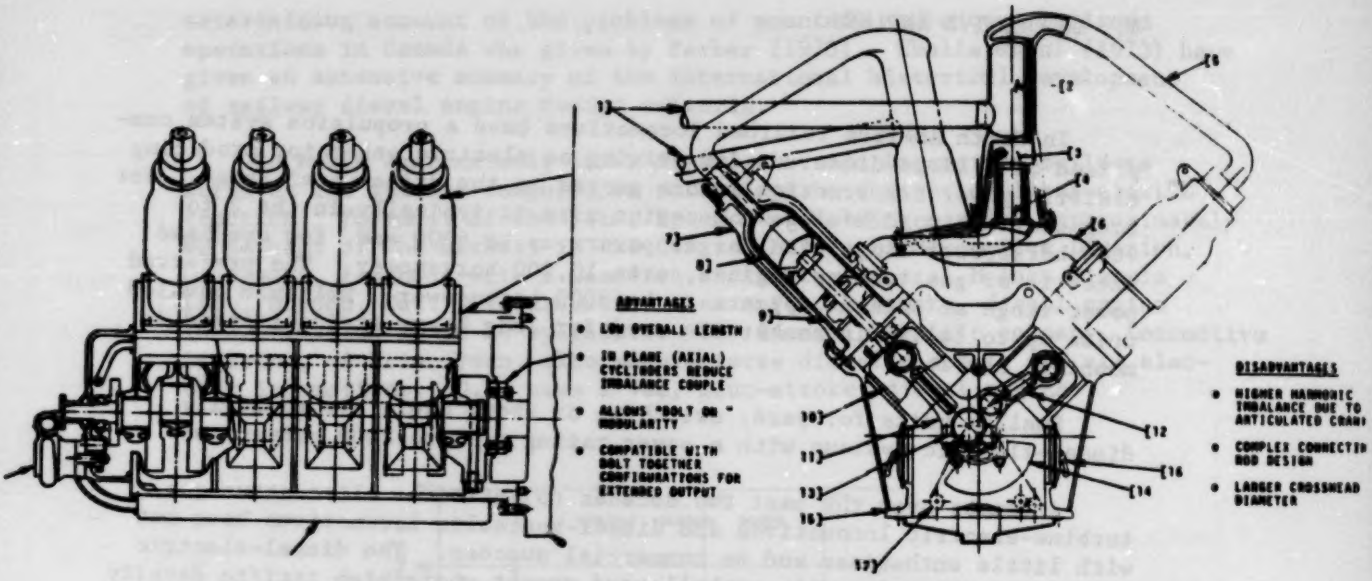

ADVANTAGES
• LOW OVERALL LENGTH
• IN PLANE (AXIAL) CYLINDERS REDUCE IMBALANCE COUPLE
• ALLOWS "BOLT ON" MODULARITY
• COMPATIBLE WITH BOLT TOGETHER CONFIGURATIONS FOR EXTENDED OUTPUT
DISADVANTAGES
• HIGHER HARMONIC IMBALANCE DUE TO ARTICULATED CRANK
• COMPLEX CONNECTING ROD DESIGN
• LARGER CROSSHEAD DIAMETER

Figure 17 Cross-Section of Articulated Vee Engine Selected by General Electric for Four-Cylinder Siemens Stirling Engine of 125 Horsepower per Cylinder Capacity, (after G.E. 1980).

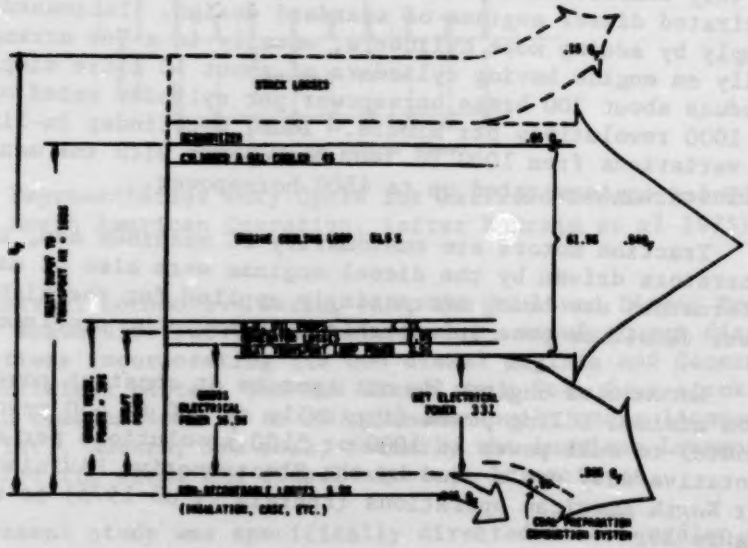


Figure 18 Energy Flow or Sankey Diagram for the Stirling Engine System, (after G.E. 1980).

4. LOCOMOTIVE ENGINES

In North America railroad locomotives have a propulsion system comprised of a large diesel engine driving an electric generator producing electric power for traction motors geared to the drive wheel axles. For long distance road haulage the engine size is typically in the 2 to 4000 horsepower range with larger powers up to 6000 and, for overhead electric or gas turbine engines, even 10,000 horsepower. The preferred power range at present appears to be 3000 horsepower. Multiple coupled units up to twelve in number are used for long heavy trains engaged in mountain service.

Smaller units for yard, switching or short hauls are typically diesel-electric systems with a power rating of 1 to 2000 horsepower.

Attempts over the past two decades to introduce alternative gas turbine-electric locomotives and diesel-hydraulic locomotives have met with little enthusiasm and no commercial success. The diesel-electric system prevails virtually unchallenged except where high traffic density makes electrification of the line and the use of electric locomotives commercially viable.

Diesel electric equipment has been developed to the stage of immense reliability. Intervals of 500,000 miles between major maintenance and locomotive lifetimes of 20/25 years are routine, both expected and demanded by the railroad operating companies. This is achieved by the use of very conservatively rated slow running turbo-charged or naturally aspirated diesel engines of standard design. Increased power is achieved simply by adding more cylinders, usually in a Vee arrangement. Typically an engine having cylinders of about 10 litre displacement will produce about 200 brake horsepower per cylinder rated output at a speed of 1000 revolutions per minute. Thus, 6-cylinder in-line engines come in variations from 1000 to 1400 horsepower with the maximum 18 or 20 cylinder engines rated up to 4500 horsepower.

Traction motors are customarily DC machines and, traditionally, the generators driven by the diesel engines were also DC machines. A.C. alternators are being increasingly applied for the higher powers where power densities pose insuperable DC commutator problems.

Locomotive engines do not operate at constant power output but vary from minimum idling power (typically at 400 to 450 revolutions per minute) to full power at 1000 or 1100 revolutions per minute. A representative duty cycle used by the Electromotive Division of General Motors for North American operations (Ephraim *et al* 1975) is reproduced in Figure 19.

An excellent survey of diesel electric railroad operations in North America was given by Ephraim *et al* (1975). A very extensive and highly

entertaining account of the problems of mountain and winter railroad operations in Canada was given by Parker (1973). Bhalla *et al* (1973) have given an extensive summary of the international historical development of railway diesel engine design criteria.

In North America only three diesel electric locomotive builders survived the intense competition following the change to diesel-elctric traction. Foremost of the trio, by an overwhelming margin, (approximately 80 percent of the market) is the General Motors Electromotive Division, La Grange, Illinois with a Canadian affiliate, General Motors Diesels Ltd., London, Ont. Electromotive uses a Vee, 2-stroke, diesel engine in combination up to 20 cylinders. The General Electric company, Locomotive Division, of Erie, Penn. also manufactures diesel-electric and all electric locomotives. G.E. uses a Vee, four-stroke, diesel engine.

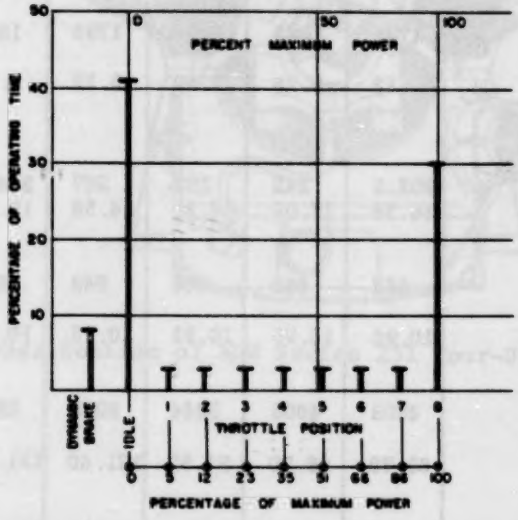


Figure 19 Representative Duty Cycle for Railroad Locomotive Engine in North American Operation, (after Ephraim *et al* 1975).

The Montreal Locomotive Works, MLW, the Rail and Diesel Products Division of Bombardier Inc., Montreal, Quebec, manufactures diesel electric locomotives incorporating its own diesel engines and General Electric electrical parts. The MLW diesel is a Vee, four-stroke, engine available in combinations up to 18 cylinders built under license from ALCO Power Inc., Auburn, New York, formerly the American Locomotive Co. well known for its steam and diesel electric locomotives.

The present study was specifically directed to a Canadian context and for this reason attention was concentrated on the MLW diesel unit. Figure 20 is a cross-section of the current production MLW Model 251 Vee diesel engine and Table 4 is a summary of the Model 251 series characteristics and ratings. Figure 21 shows the arrangement and installation envelope of a 16-cylinder engine with some additional data about

ENGINE CHARACTERISTICS AND RATINGS

NO. OF CYLINDERS	6		8	12		16		
SERIES	B	D	E	C	E	B	C	E
Horsepower (Gross)	1050	1350	1700	2150	2600	2600	3050	3300
CV (Gross)	1065	1370	1723	2180	2635	2635	3090	3340
HP/Cylinder	175	225	212.5	179.2	216.7	162.5	190.6	206.6
CV/Cylinder	177.4	227	215.4	181.7	219.6	164.8	193.2	208.8
RPM	1025	1100	1000	1025	1050	1000	1050	1100
Piston Speed (fpm	1795	1925	1750	1795	1838	1750	1838	1925
Piston Speed (M/S	9.12	9.78	8.89	9.12	9.34	8.89	9.34	9.78
Brake Mean Effective Pressure, psi	202.5	242	252	207	244.6	192.5	215	222
Brake Mean Effective Pressure, Kg/cm^2	14.24	17.02	17.20	14.56	17.20	13.54	15.12	15.63
Swept Vol./Cylinder in^3	668	668	668	668	668	668	668	668
Swept Vol./Cylinder litres	10.95	10.95	10.95	10.95	10.95	10.95	10.95	10.95
Total Swept Vol. in^3	4008	4008	5344	8016	8016	10688	10688	10688
Total Swept Vol. litres	65.70	65.70	87.59	131.40	131.40	175.20	175.20	175.20
Weight, Engine Alone, Dry lbs.	22100	24500	25670	32300	36800	40800	40800	41750
Weight, Engine Alone, Dry Kg.	10025	11113	11644	14651	16692	18507	18507	18938
Compression Ratio	12.5:1	12.5:1	11.5:1	12.5:1	11.5:1	12.5:1	12.5:1	11.5:1
Fuel Pump Plunger Diameter inches	.59	.67	.67	.59	.67	.59	.63	.63
Fuel Pump Plunger Diameter mm	15	17	17	15	17	15	16	16

NOTE: The above ratings are based on sea level, 29.92 in. Hg. (760 mm.), 60°F intake temperature (16°C) and fuel of 19,500 BTU/lb. higher calorific value (10,850 Cal. Kg.) meeting ASTM 2-D specification.

TABLE 4

Engine Characteristics and Ratings for Bombardier/MLW Series 251 Diesel Locomotive Engines

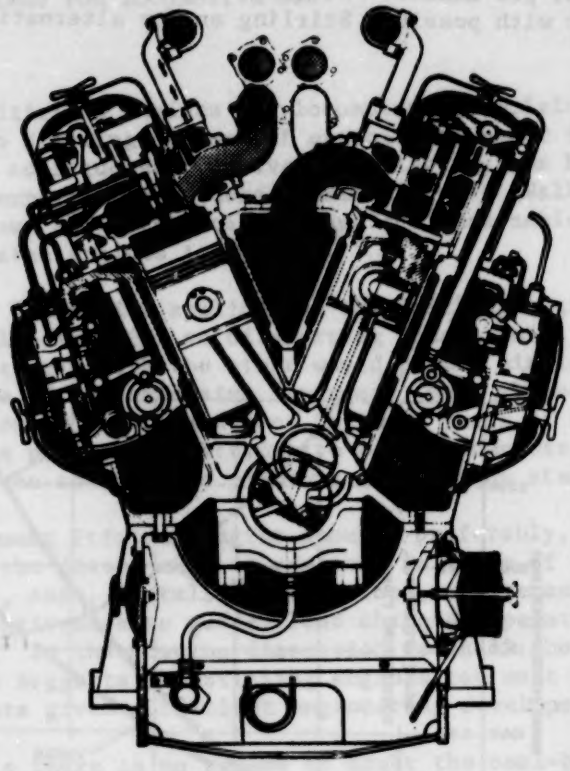


Figure 20 Cross-Section of MLW Series 251 Four-Stroke Diesel Vee Engine.

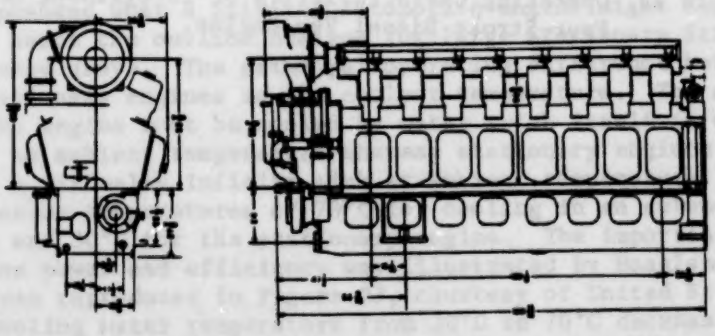


Figure 21 Arrangement and Installation Envelope of 16-Cylinder MLW Series 251-C.

the engine. Figure 22 shows the operating characteristics of the 16-cylinder series 251-C engine producing 3050 brake horsepower at 1050 revolutions per minute. We adopt this as the reference engine for comparison with possible Stirling engine alternatives.

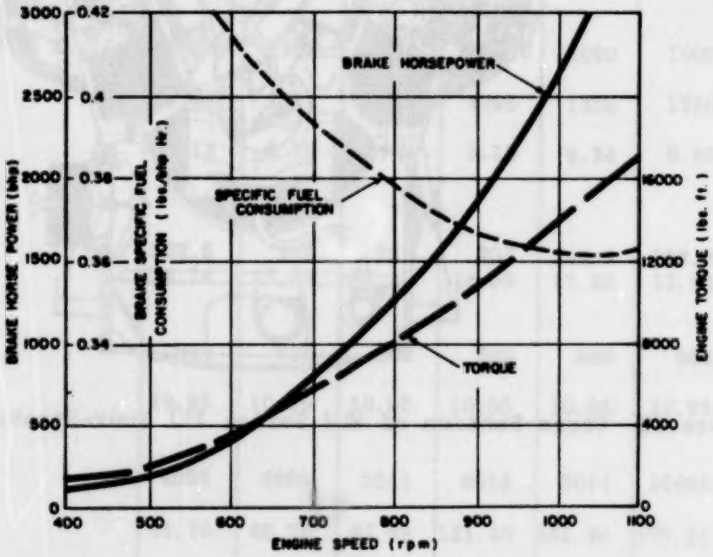


Figure 22 Operating Characteristics of 16-Cylinder MLW Series 251-C Four-Stroke Diesel Vee Engine.

5. STIRLING ENGINES FOR LOCOMOTIVE USE

The use of Stirling engines for locomotive propulsion is without precedence for no Stirling engines of appropriate power (up to 5000 horsepower) have been constructed. However, smaller engines have been used in a variety of experimental trucks, buses and cars (Walker 1980) and sufficient is known about the scaling laws for reasonable first order projections of large engines to be made.

The obvious locomotive application for Stirling engines is as a coal-burning replacement for an oil-burning diesel unit. Direct replacement of one power unit with the other would permit simple modification and continued use of the extensive inventory (many thousands) of diesel locomotives presently in use and good for many years of service. This is clearly a preferable alternative to the complete replacement of all the locomotives as with the previous change from steam to diesel.

The replacement Stirling engine should, preferably, be as good if not better than the diesel unit it replaces in terms of power, weight, size, efficiency, cost, installation envelope, maintenance cost and requirements, responsiveness to sudden load changes, operating characteristics and reliability. In the ensuing discussion we shall show that information available suggests the Stirling engines can meet most if not all these requirements given sufficient engineering development.

In principle there is no reason to doubt the coal-burning Stirling engine could become a reality given the will and determination to bring it about. The novel concepts for Ringbom engines presented herein even allow for the use of much existing locomotive diesel engine hardware and also for such non-exotic working fluids as air with high pressure, water injection for power boost at peak power operation and water as the engine lubricant.

To comprehend what a Stirling locomotive engine might look like we shall first adapt the outline designs for large stationary Stirling engines discussed above. The principal operating difference between locomotive and stationary engines is the coolant temperature. The automotive or locomotive engine must be cooled by water which itself is cooled in 'radiators' to ambient temperature whereas stationary engines are often cooled from a virtually infinite sink at ambient temperature. It is customary to assume temperatures of 70°C for cooling in an automotive/locomotive engine and 30°C for the stationary engine. The important effect this has on engine power and efficiency was illustrated by Hoagland *et al* (1978) in the diagram reproduced in Figure 23, courtesy of United Stirling. A change in cooling water temperature from 30°C to 70°C decreases engine power by 18% and decreases the efficiency by 13%. In similar fashion, a change in heater temperature affects both the engine power and efficiency. Figure 24 shows the increase in efficiency and power with increase in heater temperature. An increase from 720°C to 800°C results in an increased power of 12% and efficiency by 10%.

Another important difference between stationary and automotive engines,

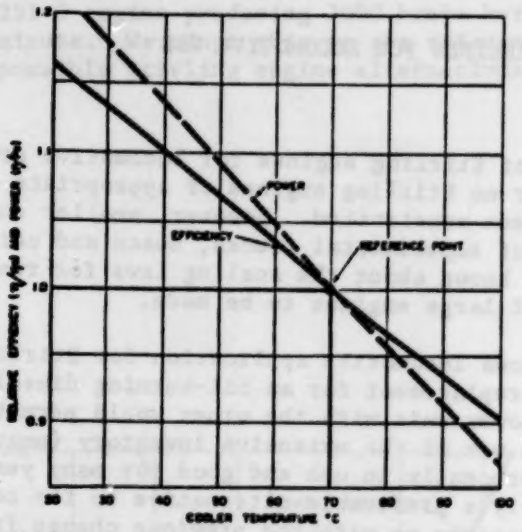


Figure 23 Effect of Cooling Water Temperature on the Power and Efficiency of a Stirling Engine, (courtesy United Stirling).

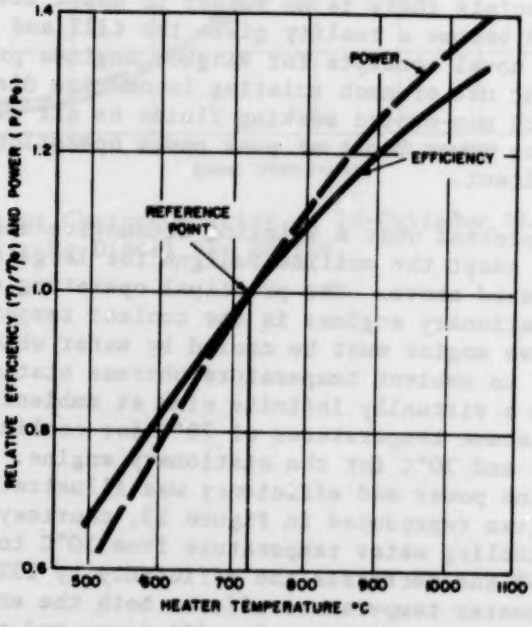


Figure 24 Effect of Heater Head Temperature on Power and Efficiency of Stirling Engines, (courtesy United Stirling).

also related to cooling, is that the cooling fan of the automotive/locomotive engine may consume a significant fraction of the engine output. Hoagland *et al* (1978) included the relationships reproduced in Figure 25 showing the fan power consumption in terms of percentage of engine output as a function of the air - water inlet temperature difference and radiator/engine size. An allowance for fan power consumption of 4% of the engine output would perhaps be realistic.

With these order-of-magnitude correctors we are able to compare existing outline designs for large stationary Stirling engines, adjusted for

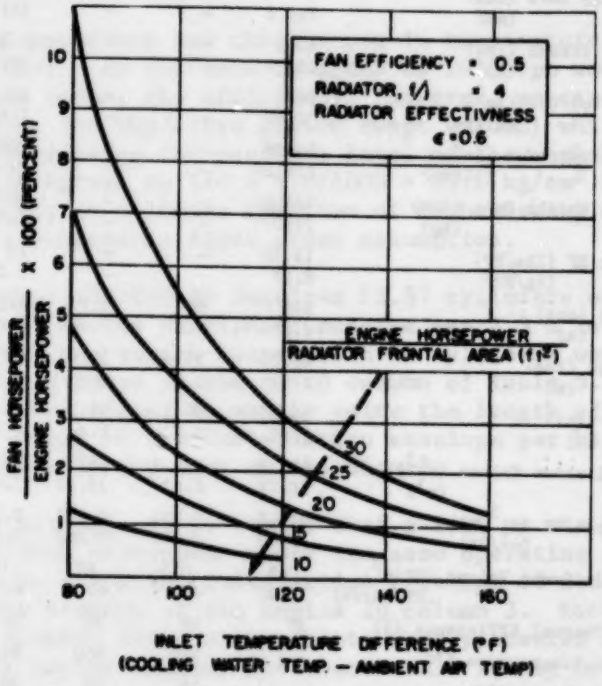


Figure 25 Cooling Fan Power Consumption as a Function of the Air/Water Inlet Temperature Difference and the Radiator Size.

automotive operation, with the reference Series M251 diesel engine. The appropriate data is summarized in Table 5. The first column entry is simply a restatement of the data for the 16-cylinder 3000 horsepower Series M251 diesel given in Table 4.

5.1 The Meijer Rhombic Engine

The second entry is the data presented by Meijer (1970) for a four-cylinder 900 horsepower stationary engine given in Figures 11 and 12 herein. For our comparison we have selected the middle range case B shown on Figures 11 and 12. The engine would be similar in arrangement to that shown in Figure 5. We must first correct the power rating to allow for operation at a heater temperature of 800°C instead of 700°C. This is the upper temperature limit for contemporary practice with sodium heat pipes and, according to Figure 24 allows an increase in the engine power and efficiency

Column No.	1	2	3	4	5	6	7
	Series M251 Diesel	Meijer Rhombic Drive Engine B-Fig. 11	Revised Meijer Rhombic	Hoagland *et al* (1978) Engine	Revised Hoagland Engine	General Electric (1980)	Revised General Electric
Horsepower (gross)	3050	900	3094	2000	3000	1000	3000
No Cylinders	16	4	14	16	16	8	16
HP/Cylinder	190.6	225	221	125	187.5	125	187.5
Speed RPM	1050	810	1000	850	1000	1200	1053
Cylinder Bore (ins)	9	-	-	7.18	9.55	7	7.97
(mm)	*228.5*	-	-	*182*	*242.5*	*177.8*	*202.4*
Piston Stroke (ins)	10.5	-	-	4.0	5.32	4.0	4.55
(mm)	*266.7*	-	-	*102*	*135.1*	*101.6*	*115.5*
Swept Vol/Cyl (in^3)	668	-	-	162	380	153.94	227
(litre)	*10.95*	-	-	*2.65*	*6.22*	*2.52*	*3.72*
Total Swept Vol (in^3)	10688	-	-	2592	6080	1231	3632
(litre)	*175.20*	-	-	*42.4*	*99.62*	*20.16*	*59.5*
Engine Weight Drag (lbs)	40800	-	-	32000	39336	12222	36600
(kg)	*18507*	-	-	*14500*	*17880*	*5555*	*16636*
Weight/HP (lbs/HP)	13.37	-	-	16	13.1	12.2	12.2
(kg/HP)	*6.06*	-	-	*7.25*	*7.25*	*5.55*	*5.55*
Length (ins)	191	90	317	143	164	72	166.5
(m)	*4.89*	*2.3*	*8.05*	*3.632*	*4.16*	*1.82*	*4.22*
Breadth (ins)	60.5	39.4	39.4	50	57.5	84	96.6
(m)	*1.55*	*1.0*	*1.0*	*1.270*	*1.45*	*2.13*	*2.45*
Depth (ins)	96.5	94.5	94.5	68	78.2	78	89.7
(m)	*2.47*	*2.4*	*2.4*	*1.727*	*1.98*	*1.37*	*2.27*
Installation Volume (ft^3)	645.3	193.9	683	281.4	421.8	273	834.9
(m^3)	*18.75*	*5.52*	*19.32*	*7.96*	*11.94*	*5.31*	*23.46*
Vol. per HP (ft^3/HP)	0.215	0.215	0.215	0.1406	0.1406	0.273	0.273
(m^3/HP)	*.0061*	*0.006*	*0.006*	*0.0039*	*0.0039*	*0.0053*	*0.00782*
HP/Unit Swept Volume (HP/m^3)	.285	.655	.655	.7716	.493	.813	-
(HP/litre)	*17.40*	*40*	*40*	*47.16*	*30.14*	*49.58*	*50.4*
Max. Thermal Efficiency (%)	37	39	39	39.6	35	40	40
Heater Temperature (°C)	-	700	800	800	800	760	800
Cooling Water (°C)	-	25	70	30	70	80	70
Working Gas	Air/Fuel	Helium	Helium	Helium	Helium	Helium	Helium
Mean Operating Pressure (ps)	-	1552	1261	2030	1297	1470	1470
(MPa)	-	*10.7*	*8.7*	*14.00*	*8.94*	*10.1*	*10.1*
Max. Pressure (psi)	1450	-	-	-	1.733	1984	1984
(MPa)	*1.0*	-	-	-	*11.95*	*13.68*	*13.68*
Min. Pressure (psi)	-	-	-	-	866	955	955
(MPa)	-	-	-	-	*5.97*	*6.58*	*6.58*
Mean Eff. Pressure (psi)	215	-	-	-	-	-	-
(MPa)	*1.48*	-	-	-	-	-	-
Engine Type	Diesel	Stirl. Rhomb.	Stirl. Rhomb.	Stirl. Siem.	Stirl. Siem.	Stirl. Siem.	Stirl. Siem.
Column No.	1	2	3	4	5	6	7

TABLE 5

Comparative Data for 3000 Horsepower Diesel and Stirling Locomotive Engines

of 18% and 16%, respectively. The increase in cooling water temperature from 25°C to 70°C imposes penalties on the power and efficiency according to Figure 23 of 20% and 15%, respectively. Therefore the HP/cylinder at 800°C upper temperature and 70% cooling temperature would be:

$$\text{HP/cylinder} = 225 \times 1.18 \times \frac{1}{1.20} = 221 \text{ HP/cylinder}$$

and the maximum efficiency under these conditions will be

$$\eta_{max} = 39 \times 1.16 \times \frac{1}{1.15} = 39\%$$

The adjustments, necessary for the changes in temperature, virtually compensate for each other. An increase in speed to 1000 rpm would increase the power density and reduce the efficiency. However, we can remain at the same power density (40 bhp/litre piston swept volume) while increasing speed to 1000 rpm by decreasing the pressure level of the working fluid from a mean value of 110 kg/cm^2 to 110 x 810/1000 = 89.1 kg/cm^2. This assumes the engine power output is a linear function of both working fluid pressure and engine speed - a reasonable first order assumption.

A power output of 3000 HP requires 13.57 cylinders of 221 HP/cylinder, say 14 cylinders. If the 4-cylinder machine was 2.3 m long, i.e. 0.595 m per cylinder, then the engine length with 14 cylinders would be 8.05 m. These figures are entered in the third column of Table 5. It should be noted this monstrous engine is nearly twice the length of the locomotive diesel but the volume of the installation envelope per horsepower at 0.006 m^3/hp is *exactly the same as the diesel.*

In a practical case it is likely that a four or six cylinder rhombic drive engine of 3000 horsepower under the same operating conditions could be accommodated in a smaller installation envelope of about half the length and 1.5 times the breadth of the engine in column 3. Exceptionally compact opposed piston rhombic drive arrangements were pioneered at Philips in the early 1960's for marine engines and these could likely be adapted for locomotive use.

In their later study, General Electric found a number of advantageous features in rhombic drive engines but eventually discarded them because they perceived seal maintenance requirements deemed to be unacceptable.

5.2 The Hoagland Siemens Engine

The next entry, in column 4 of Table 5, is the specification given in Table 1 for the representative 2000 horsepower engine devised by Hoagland *et al* (1978) by scaling the United Stirling P75 engine.

We wish to extend to process to determine the parameters for a 3000 horsepower P75-type engine. First we note the Hoagland engine assumed a cooling temperature of 30°C instead of the 70°C for locomotive operation. Adjustment of the cooling temperature to 70°C involves a penalty of 18 percent and 13% on the power and efficiency, respectively. Therefore the engine efficiency is 39.6/1.13 = 35% and the power per cylinder is 125/1.18 = 105.93 BHP. If we maintain the same 16 cylinders of 4 modules of 4-cylinder

engines in an 8-cylinder in-line, twin crank arrangement then the power output required is 3000/16 = 187.5 BHP. Since the power varies as the scale factor γ squared then to scale to 187.5 BHP, the scale factor becomes: $\gamma = \sqrt{187.5/105.9} = 1.33$, and the bore, stroke and speed scale as follows:

Bore = 7.18 x 1.33 = 9.55 ins.
Stroke = 4.0 x 1.33 = 5.32 ins.
Speed = 850/1.33 = 639 rpm.

For comparability with the diesel we prefer the speed to be 1000 rpm and to keep the same power density and thermal efficiency we shall compensate for the increase in speed by decreasing the mean working gas pressure from 2030 lb. per sq. in. to 2030 x 639/1000 = 1297 psi. The reduction in working fluid pressure will almost certainly reduce the engine weight per horsepower perhaps by half the pressure reduction factor, i.e.

$$\frac{16}{2030} \times (2030 - \frac{1}{2}(2030 - 1297) = 13.11 \text{ lbs. per HP}$$

and the total weight of the engine then becomes 39336 lbs.

The volume per horsepower of the installation envelope will likely remain about the same at 0.1405 ft^3/horsepower so the total volume is 421.8 cu. ft. Assuming the geometry of the engine expands in the same proportion (i.e. $\sqrt[3]{1.5}$) then the length, breadth and depth become:

Length = 143 x 1.15 = 164 ins.
Breadth = 50 x 1.15 = 57.5 ins.
Depth = 68 x 1.15 = 78.2 ins.

Column 4, the revised Hoagland engine may now be compared with column 1, the contemporary diesel locomotive. The Stirling engine is smaller, lighter and marginally less efficient than the diesel engine of the same power running at the same speed. The stroke of the Stirling engine is only about half that of the diesel engine although the cylinder diameters are about the same. The mean operating pressure of the Stirling engine is 1297 lbs. per sq. in. (say 1300) and if the pressure ratio, p_{max}/p_{min}, is 2:1, the maximum and minimum pressures are 1733 and 866 lbs. per sq. in.

We do not know the maximum pressure in the diesel engine. The specification shows the brake mean effective pressure to be 215 lb. per sq. in. and according to Bhalla (1973) a b.m.e.p. of 215 lb/sq. in. (15.12 kg/cm^2) in a diesel engine corresponds to a maximum cylinder pressure of 102 kg/cm^2, or 1450 lb. per sq. in. The pressure regimen in the diesel is therefore somewhat less than that of the Stirling but the rate of *pressure change* during combustion in the diesel is very much higher (several orders of magnitude) than the highest rate of pressure change in the Stirling engine. The pressure-time characteristic in a Stirling is sinusoidal to a first approximation.

5.3 The General Electric Vee Engine

Column 6 of Table 5 contains entries for the 1000 horsepower Vee two-piston machine gleaned from the General Electric 1978 report. This engine

of 8 cylinders is rated at 1000 horsepower. If we use 16 cylinders as in most of the other engines we could simply have two modules of the 8 cylinder engines for a total 2000 horsepower engine.

The heater and cooler temperatures are lower (760 instead of 800°C) for the heater and higher (80 instead of 70°C) for the cooler. Therefore we will adjust the power output to our reference condition. Increasing the heater temperature from 760 to 800°C increases the power and efficiency by 7 and 6 percent, respectively. Decreasing the cooler temperature from 80 to 70°C increases the power and efficiency by 8 and 6 percent, respectively. Therefore applying these temperature corrections: $P = 125 \times 1.07 \times 1.08 = 144.45$ hp/cylinder. For a 3000 horsepower 16-cylinder engine we need $3000/16 = 187.5$ hp/cylinder so the scale factor γ is: $\gamma = \sqrt{187.5/144.45} = 1.139$. The revised bore, stroke and speed are:

Bore = 7 x 1.139 = 7.97 ins.
Stroke = 4 x 1.139 = 4.55 ins.
Speed = 1200/1.139 = 1053 rpm

If we assume the length, breadth and depth of the engine grow in the same proportion (i.e. $\sqrt[3]{1.5}$) while maintaining the same volume/horsepower of the installation envelope we can estimate the overall size to be:

Length = (2 x 72) x 1.15 = 166.5 ins.
Breadth = 84 x 1.15 = 96.6 ins.
Depth = 78 x 1.15 = 89.7 ins.

Maintaining the same weight per horsepower, we estimate the weight of the 2000 horsepower engine to be 36,600 lbs.

Column 7 of Table 5 for the revised General Electric Vee Stirling engine may now be compared with column 1 for the MLW Series 251-C diesel. The Stirling engine is lighter but the installation envelope is substantially greater than the diesel engine. This is not unexpected for on the engine cross-section diagram given in Figure 17 there seems to have been little attempt to produce a tight compact design. Revised design of the heat pipe arrangement could almost certainly produce a substantial reduction in the installation envelope, a significant reduction in the height. However, there seems to be little opportunity to reduce the engine breadth (at 96.6 ins., 50 percent greater than the diesel at 60 ins.). This high breadth arises from the choice of a 90° Vee (to get the proper piston phasing) and the very long cylinders that result from the use of a crosshead, seal unit and a Heylandt crowned piston.

5.4 Heat Source

In comparing the Stirling engines of column 5 and 7 with the diesel of column 1 in Table 5, it is necessary to bear in mind that the heat source for the Stirling engine is not included in the installation envelope. The Meijer rhombic drive engine of column 2 and in revised form in column 3 did include the heater head and exhaust gas/inlet air preheater. Appreciable savings in the installation envelope (up to 40 percent) would follow by deleting the exhaust inlet air preheater and heat source from the rhombic engine.

Physical separation of the heat source from the engine with heat pipe coupling is advantageous for it provides considerable flexibility in location of the heat source. For the engines of interest here a fluidized bed coal combustor is the most likely heat source.

The energy flow diagram shown in Figure 16 and devised by General Electric indicates the gross mechanical power output to be 39% or the engine heat input reckoned to be 80 percent of the fluidized bed input. Therefore 3000 horsepower engine output would require an energy input to the bed of 3000 x 0.746 x 1/0.39 x 1/0.8 = 7173 kW.

Dunn *et al* (1975) indicates the maximum power density of fluidized bed combustors to be 650 kW/m^3. Therefore, a fluidized bed combustor of 7173/650 = 11 m^3 would be associated with a 3000 horsepower Stirling engine. The bed must also contain the heat pipe boiling sections so that some additional volume of bed would be necessary, say, a total of 12 m^3 or 4 m^3 per 1000 horsepower.

In the Stirling powered locomotive the fluidized bed combustor would most likely be accommodated directly under the engine between the trucks (bogies) beneath the frame.

The MLW M630 diesel electric locomotive has a space between trucks of approximately 6.76 m x 2.56 m x 1.08 m or a volume of 18.71 m^3, 50 percent more than volume required for the fluidized beds. The remaining volume could be used for buffer coal storage.

5.5 Heat Transport

Heat from the fluidized bed combustor is conveyed to the engine by a heat pipe or by convective gas transport. All the studies referenced above include extensive discussions of the different systems and recommend the sodium heat pipe for convenience, cost and reliability, notwithstanding the slight safety hazard resulting from accidental rupture of the heat pipe and the uncontrolled release of the highly reactive sodium metal. Convective gas transport loops consume substantial power to move the gas. Direct fired grates with radiation heating or immersion of the heater tubes in the fluidized bed offer great simplicity but unfortunately it is not possible to devise these systems to maintain the same maximum temperature throughout; there will always be 'hot spots'. Consequently, the *mean* temperature must always be 50 to 100°C less than the metallurgical limit to allow for the hot spots to still below the critical temperature. With a heat pipe all the condensing surfaces are at the same temperature (to a degree or two) and so the mean temperature is also the maximum temperature. We have seen above the considerable (10-12 percent) improvements in power and efficiency that result from increasing the heater temperature from 700 to 800°C.

Another, and equally significant, advantage of the heat pipe is the very high rates of heat transfer during condensation of the sodium vapour on the engine heater tubes, much greater than could be obtained by radiation heating or immersion in the fluidized bed. As a result, the heater tubes can be decreased in length and diameter with a very considerable advantage to the engine performance arising from the reduction in internal void volume, the so-called 'dead space'.

Dunn *et al* (1975) indicates heat transfer rates of 100 kW/m^2 of heat transfer surface for the boiling section of the heat pipe immersed in the fluidized bed. Similarly, heat fluxes of 15 kW/cm^2 at temperatures above 700°C may be gained. The A.M.T. (1980) report contains substantial data on heat pipe design for Stirling engine application and suggests the present size limit of sodium heat pipe technology is about 250 kW per heat pipe. The G.E. (1980) and Hoagland *et al* (1978) reports also contain much valuable information concerned with fluidized bed combustion and sodium heat pipe design and application. Another substantial source of information concerning the application of heat pipes to Stirling engines is a report from the Thermo-Electron Corp., Waltham, Mass. (Morgan 1978) concerned with thermal energy storage for Stirling locomotive engines.

The only practical experience of a Stirling engine operating on a coal-fired fluidized bed with sodium heat pipe coupling was reported by Asselman (1976) using a single-cylinder 10 horsepower Philips Stirling engine. Dunn *et al* (1975) have reported experiments with a small (less than 1 horsepower) Stirling engine energized by a sodium heat pipe from a propane fuelled fluidized bed.

5.6 Cost

Reliable data on the cost of Stirling engines is simply non-existent. Apart from the cryocooler use, no commercial applications have evolved in quantities sufficient to establish reasonable continuous production. No large locomotive-size Stirling engines have ever been constructed.

All the estimates available appear to agree that the capital cost of Stirling engines will be greater than for the diesel engine of the same power. For large engines Hoagland *et al* (1978) suggests ... *'In production, stationary Stirling engines burning distillate fuels are likely to be 25-50% more expensive than diesel engines of similar size'* and also *'Stirling engines operating on solid fuels (coal) are likely to be 30-50% more expensive than Stirling engines running on distillate fuels'*.

Combining these two effects we gain the limiting range:

Lower limit = 12.5 x 1.3 = 1.62
Upper limit = 1.5 x 1.5 = 2.25.

That is, coal-burning Stirling engines would likely be 1.6 to 2.25 times the capital cost of a diesel engine of the same power. Of course the overall life cost includes the operating (fuel and lubricating oil) costs, the maintenance and repair costs and amortization. Sufficient data to allow a meaningful comparison of total life cost is not available but there is every reason to contemplate the same very long life for Stirling engines that is routinely gained and expected from diesel engines (500,000 miles between major overhauls). Stirling engines have no valves and the combustion products are not in contact with the moving parts, therefore lubricant oil consumption is virtually zero and the wear rates of rubbing surfaces is very low.

In their study of large Stirling engines, General Electric (1980) estimated costs of $230 to $297 per horsepower for a 1000 horsepower engine at a production rate of 1000 engines per year. For comparison they quoted the cost for a 2000 horsepower locomotive engine of $117 per horsepower. The Stirling engine was more expensive by a factor of 1.95 to 2.5 times the diesel, thereby confirming the independently derived Hoagland figures quoted above.

5.7 Impediments to Application

At present there is no incentive for manufacturers or operators to consider replacing diesel engines with Stirling engines. There is no shortage of diesel fuel and although substantial price increases have been sustained there is not yet an economic imperative for an alternative. Stirling engines operating on diesel fuel could be developed but presently they do not exist in very large sizes so the cost of research and development would be high. The capital cost in production would be 1 1/2 to 2 times the cost of the diesel and the only real advantage of the Stirling engine would be the low noise level.

As diesel fuel prices rise and become restricted in availability the impetus for a coal-burning replacement will become more pronounced provided that coal prices remain at their present comparatively low level. A difference in capital cost of the engine of two or even three to one becomes relatively insignificant in the lifetime cost if the expensive engines can operate on a cheaper fuel.

Of course the steam engine will be in contention to replace the diesel but there seems to be no easy way to fit a steam engine of the same power and efficiency into the installation envelope of a diesel unit. The reverse conversion to steam would require a whole new fleet of locomotives as the earlier conversion did in the 1950's.

We have seen from the data in Table 5 above, the Stirling engines do appear to offer the real possibility of an alternative coal-burning, silent, engine able to replace the diesel in the same space, produce the same power at the same efficiency with the prospects of high reliability. This would permit gradual conversion of existing fleets of oil-burning diesel locomotives to coal-burning Stirling locomotives. Of course the conversion cost would not be trivial, but the savings in converting a diesel locomotive rather than replacing it would likely be enormous, savings of perhaps half the cost of the locomotive.

Despite all these advantages the prospects for the large locomotive Stirling engines of the type discussed above are not very bright. The reason is, in a word, 'seals', the Achilles heel of all modern Stirling engines of conventional design.

To justify the level of concern about the seals problem we include here, in Appendix 6, the relatively extensive notes on piston and rod seals given in Hoagland (1978) and the General Electric (1980) reports on large stationary engines.

Present designs of displacer rod seals serve the dual function of

preventing the egress of working fluid (helium) from the working space and the ingress of lubricating oil to the working space. Leakage of the working fluid can be overcome by periodic recharging, a tedious, expensive, but, perhaps tolerable, process. Leakage of oil in the working space is more serious for the oil blocks the fine interstices in the regenerator and ultimately carbonizes. This increases the regenerator pressure drop and decreases the power and efficiency of the engine. Cleaning the internals of an engine requires something considerably more than 'routine maintenance'.

Another critical problem arises with the dry-rubbing seals of the piston/displacer in Siemens-Stirling engines. Almost invariably there is a preferential leakage from one Stirling system to the adjacent, interconnected systems with the consequence that one will be starved of fluid while the other will be overcharged.

Research on seals for Stirling engines over the past 3 decades has consumed a king's ransom of research and development effort, yet recently Hoagland *et al* (1978) wrote that *'Using state-of-the-art seals, United Stirling obtains leakage of hydrogen from the 4-piston rod seals of a 100 brake horsepower, 4-cylinder, double-acting engine ranging from 0.1 to 0.2 grams/hour at full load. The life of their piston rings, as well as rod seals is currently about 4000 hours'.*

Compare this with the life of 500,000 miles expected from a railroad locomotive. Even allowing such a high mean speed of 20 miles an hour, the life is then 500,000/20 = 25,000 hours, six times the present seal life after all the intense and prolonged development effort.

The use of helium instead of hydrogen may permit an extended life, but even twice the life, 8000 hours, is simply unacceptable.

Equally unpromising after twenty years of development is the Philips roll sock seal, hailed at the time of its reinvention in the 1950's (following the initial invention by Leonardo da Vinci much earlier) as the break-through resolving the Stirling engine seal problem. Presently available materials and designs for rolling diaphragm seals simply do not have the necessary durability and are too complicated with excessive maintenance requirements. Currently NASA and their contractors have substantial alternative seal development work in progress but nothing that appears even remotely possible to attain the lifetime necessary for the Stirling engine to gain acceptance.

In this desperate situation new approaches are clearly required. We present those for consideration in the next section.

6. NOVEL APPROACHES: THE CONVERSION OF DIESELS TO STIRLING ENGINES

In the preceding section we saw how large coal-burning Stirling engines could be designed and constructed to replace oil-fuelled diesel engines in railroad locomotives and so permit the gradual conversion and continued usage of existing locomotive fleets. The greatest impediment to this occurring in practice is a) the limited life of the rod seals used to keep the helium in, and the lubricant out, of the working space and b) the piston seals used to prevent the transfer of working fluid to and from the interconnected systems in double-acting Stirling engines.

We propose to overcome these problems by the approaches outlined below.

6.1 Use Single-Acting Engines

Abandon the use of double-acting engines and revert to ensembles of single-acting engines on a common crankcase. This eliminates the starvation of one system and overcharging of its adjacent system. In an assembly of single-acting systems individual members can fail without affecting the total operation of the engine.

6.2 Use Air as the Working Fluid

Use air as the working fluid instead of helium and eliminate the need for a *hermetic* gas seal. Any reasonable leakage from the working space can be simply replenished from a reservoir fed by an engine-driven air compressor.

Air was used almost universally as the working fluid in the hot air engines of the earlier age. It was used in all the early Philips engines from 1938 to 1952 until they perceived a need to attain high power densities and adopted the low molecular weight gases, helium and hydrogen, as the working fluid. At that point it was no longer appropriate to refer to 'air engines' and so the generic name 'Stirling engines' was bestowed, by Rolf Meijer of Philips, on machines of this type.

Molecular weight M and, to a lesser extent, the viscosity μ are the principal factors affecting the choice of working fluid. A simplified steady flow analysis was given by Walker (1980) to show that the ratio of pumping power P to heat transferred Q was least for hydrogen, and second best, for helium. Water was considerably less attractive and air even less so. For a given P/Q ratio the comparative heat transfer was:

Air	= 1
Water Vapour	= 1.5
Helium	= 2.5
Hydrogen	= 3.5

Hoagland *et al* (1978) found the appropriate scaling of engine speed N with different working gases for the same thermal performance, displacement and operating temperatures and pressure to be:

Gas	Molecular Weight	Viscosity at 200°F (lb_m/hr-ft)	Relative Speed N_{H_2}/N
Air	29	0.052	4.30
Steam	18	0.031	3.58
He	4	0.065	1.25
H_2	2	0.025	1.0

Further evidence for the low power density to be expected from the use of air as the working fluid was provided by Meijer with the data given in Figure 11. This is reproduced in Figure 26 with the additional characteristic for steam added by proportionate interpolation according to the results of the Hoagland analysis given above.

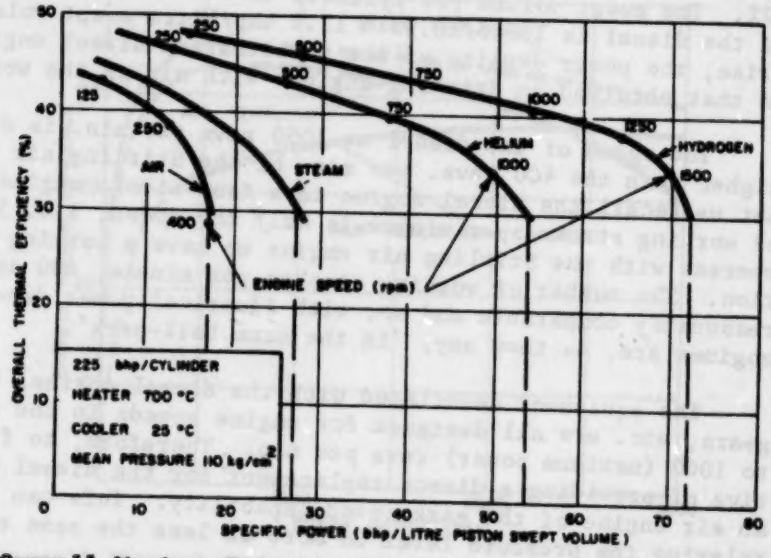


Figure 26 Overall Maximum Possible Thermal Efficiency as a Function of Engine Specific Power for Air, Helium and Hydrogen as Presented by Meijer (1970) with a Conjectural Curve for Steam Added.

We know virtually nothing of the details of the Meijer study presented but it is perhaps well to recall it was done at Philips Laboratory by a group very concerned with engines of high power density and the use of low molecular weight fluids. It is reasonable to anticipate that no special efforts were made by them to favour the use of air but rather, if anything, to demonstrate the unsuitability of air as a working fluid. Nevertheless, with faith in the integrity of Meijer and his co-workers at Philips, we shall accept at face value their results for air. A subsequent study by Michels (1976) of the small Philips 1-98 single-cylinder engine confirmed, more or less, that the thermal efficiency is relatively independent of the working fluid but the power density is much less for air than for helium or hydrogen. The reason is largely that

with air the engine must be operated at a slower speed to maintain the same efficiency as operation with the light gases.

In Figure 26 we see that at an overall thermal efficiency of 30 percent the power density with air is only 17 bhp/litre piston swept volume at a speed of approximately 400 revs per min. For steam the power density is 27 bhp per litre, for helium 54 bhp per litre at 1200 revs per minute and for hydrogen 72 bhp per litre at 1800 revs per min., all at the same 30 percent thermal efficiency. The future for air therefore, appears quite bleak for it can produce only a quarter of the power, at about a quarter the speed, of the hydrogen engine.

Now let us look at the power density of the reference diesel engine given in Table 4 and 5. The rated output is 190.6 brake horsepower per cylinder at a speed of 1050 revs per min. with a thermal efficiency of 37%. The swept volume per cylinder is 10.95 litres so the power density of the diesel is 190.6/10.95 = 17.4 bhp/litre swept volume. To our surprise, the power density of the turbocharged diesel engine is identical to that obtained in Stirling engines with air as the working fluid!

The speed of the diesel at 1050 revs per min. is of course very much higher than the 400 revs. per min. of the Stirling air engine. However, let us recall the diesel engine is a four-stroke engine so the number of working strokes per minute is *half the speed*, i.e. 525 revs per min. whereas with the Stirling air engine we have a working stroke every revolution. The number of working strokes per minute, 400 and 525, are reasonably comparable and so, with identical power density, the two engines are, as they say, 'in the same ball-park'.

The equipment associated with the diesel engine, the generator, pumps, gears, etc. are all designed for engine speeds in the range of 400 (idling) to 1000 (maximum power) revs per min. Therefore, to fulfill our objective of providing a direct replacement for the diesel engine we must have an air engine of the same speed capability. This can be achieved by relaxing the pressure level to more or less the same extent as we increase the speed.

This comes about because, to a first approximation, the power output of the Stirling engine is a linear function of both the speed and the mean pressure level. The same efficiency and power output can be maintained at *double* the speed by *halving* the pressure. That is, having the same mass rate of flow of working fluid through the active spaces of the engine. In this particular case we can increase the speed from 400 to 1000 revs per min., a factor of 2.5 times, and to compensate reduce the mean pressure by the same factor from 110 kgf/cm^2 (1550 lb. per sq. in., 10.7 MPa) to 44 kgf/cm^2 (620 lb. per sq. in., 4.28 MPa).

The pressure compression ratio (p_{max}/p_{min}) in Stirling engines with a gaseous working fluid is approximately 2:1 and never greater than 2.5:1. Here we assume a value of 2:1 for the air engine and therefore with a mean pressure of 620 lb. per sq. in. we can estimate the maximum cylinder pressure to be:

$$p_{max} = (4/3)620 = 826.6 \text{ lb. per sq. in. } (5.73 \text{ MPa})$$

and the minimum pressure to be:

$$p_{min} = 826.6/2 = 413 \text{ lb. per sq. in. } (2.86 \text{ MPa}).$$

The maximum pressure in the diesel cylinder is not given in the manufacturers data but the brake mean effective pressure (b.m.e.p.) is quoted at 215 lb. per sq. in. (15.12 kg/cm^2). Bhalla *et al* (1973) (Figure 11 of their paper) show that at this level of b.m.e.p. diesel engine peak pressures are generally in the range of 102 kg/cm^2 (1449 lb. per sq. in.).

The maximum pressure of 826 lb. per sq. in. proposed for the 'high speed', 1000 revs per min. air engine is therefore comfortably within the design capability of the diesel engine parts, cylinder walls, crankshaft, bearings, etc. It is timely to also recall that the rate of

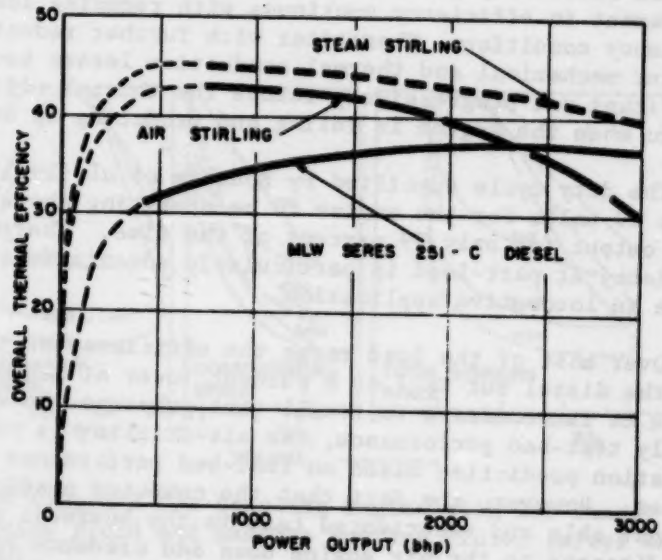


Figure 27 Overall Thermal Efficiency for a Stirling Engine with Air and Steam Working Fluids Compared with a Contemporary Locomotive Diesel Engine.

pressure change in the diesel engine during combustion is orders of magnitude greater than the maximum rate of pressure change in the Stirling engine. The latter has a slow changing, almost sinusoidal pressure-time characteristic.

Yet another factor in favour of the air engine comes from a consideration of the overall thermal efficiency as a function of the engine output, shown in Figure 27. Characteristics for the Stirling engine with air and with steam as the working fluid are compared with the reference MLW Series 251-C diesel engine. Data for the thermal efficiency of the diesel engine was drawn from the manufacturers published brake specific fuel consumption/speed characteristic (Figure 22) converted to brake thermal efficiency by assuming a fuel gross calorific value of 19,000 BTh.U./lb. Data for the air-Stirling engine was deduced from Figure 26 by

assuming the 30 percent/17 BHP/litre point, at the extremity of the characteristic for air, to be the full load condition equivalent to 3000 horsepower.

Comparison of the efficiency characteristics for the diesel and air-Stirling engines shows the diesel engine has the well known 'flat' efficiency characteristic of that type of engine increasing progressively to a maximum value near the maximum power output.

The Stirling engine also possesses a relatively flat efficiency characteristic but attains its maximum efficiency at about one-quarter full power and thereafter declines progressively with increase in load. An increase in efficiency with reduced output is typical of Stirling engines in all sizes regardless of the working fluid. It arises because the internal aerodynamic and, to a lesser extent, mechanical losses decrease relatively greater as the engine power output decreases. The improvement in efficiency continues with reducing load to the maximum efficiency condition. Thereafter with further reduction in power the constant mechanical and thermal conduction losses become increasingly significant and progressively reduce the thermal efficiency ultimately to zero when the engine is idling and producing no brake power.

The duty cycle specified by Ephraim *et al* (1975), reproduced in Figure 19 calls for the engine to be operating above 50 percent maximum power output for only 39 percent of the time. Therefore a high thermal efficiency at part-load is particularly advantageous for the Stirling engine in locomotive application.

Over most of the load range the efficiency of the Stirling is better than the diesel but is 7 or 8 percent lower at maximum power. Here of course it is necessary to recall that, whereas the diesel data is presumably test-bed performance, the air-Stirling is simply a computer simulation prediction based on test-bed performance with very much smaller engines. However, the fact that the computer predictions were made by the most able and experienced team in the business and not particularly well disposed to the air engine does add credence to their projections.

The characteristic for the steam-Stirling shown on Figure 27 was derived from that shown on Figure 25 and is nothing more substantial than speculation. Steam-Stirling engines have never been investigated in depth apart from one miniature artificial heart engine developed by the Thermo-electron Corp., Waltham, Mass., for the U.S. National Institutes of Health, discussed in some detail by Walker (1980).

6.3 Use Steam Injection for Maximum Power

Use steam injection in the hot expansion space to obtain a power boost during periods of high power operation. This corresponds to the process of water or methanol injection in aircraft internal combustion engines when taking off or in combat - Rolls Royce Merlin engines in Spitfires had this power boost feature in World War II.

In the steam injection-Stirling engine water at high pressure would be injected to the expansion space from a monotube, once through, supercritical boiler immersed in the fluidized bed or the sodium vapour heat

pipe near the expansion space. The steam would be admitted to the engine by a valve operated automatically by the control system when the displacer was approaching top-dead centre. The steam injected would condense in the regenerator, and accumulate in the liquid phase in the cold compression space. It would act as the lubricant and be pumped out of the working space to the crankcase by the piston seals. The water naturally descends the temperature gradient from the hot expansion space to the cold compression space according to the Law of Minimum Energy States. Its presence in the compression space is beneficial for it acts as the lubricant for the seals. Furthermore, it improves the efficiency of the seals since they are now *hydraulic* rather than *pneumatic*. Another advantage is that the water tends to make the processes of compression in the compression space closer to isothermal than adiabatic processes.

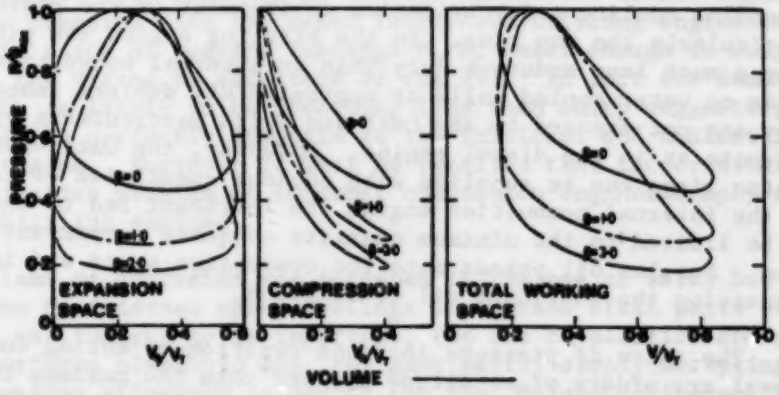


Figure 28 Work Diagrams for Stirling Engine with Two-Phase, Two-Component Working Fluid and Gaseous Working Fluid, (after Walker *et al* 1980).

Preliminary studies at the University of Calgary of two-phase two-component working fluids (air and water) in Stirling engines have shown that very substantial increases in output can be gained compared with a conventional gaseous working fluid. Figure 28 shows the theoretical work diagrams obtained by Walker *et al* (1974) for a Stirling engine with a two-phase two-component working fluid compared with a simple gaseous working fluid. Injection of significant amounts of steam during periods of high power demand could likely double the power output of the engine.

Insufficient work has been done to predict the effect of steam injection on thermal efficiency but a substantial reduction might be expected because of the 'one-way' operation of the thermal regenerator and the consequent 'enthalpy drain' to the cold side. The conjectural prediction in the earlier section that steam would actually be more efficient than air assumes that the condensing steam *returns back* through the

regenerator instead of the one-way passage of the steam injection process described here.

6.4 Use Water as the Lubricant

Another novel approach is to use water as the engine lubricant instead of oil. Oil is a problem with conventional Stirling engines because it contaminates the working space. It would be difficult indeed to keep the water circulating in the steam injection circuit from becoming contaminated with oil. All these problems can be resolved by simply eliminating oil and lubricating with water.

Water is not as good a lubricant as oil but is tolerable in special circumstances. Its use as a lubricant in oxygen compressors and large rolling mill bearings is quite routine and has been for many years. In an internal combustion engine the function of the lubricant is just as much to cool as to lubricate. This is particularly true for the very demanding cooling and lubrication environment of the piston rings, particularly the top ring. In the Stirling engine the piston sealing rings have a much less ardurous duty than an internal combustion engine. They slide on water cooled walls at approximately ambient temperature for they are not exposed to the heat and solid particulates of the combustion products as in the diesel engine. Moreover, the water-lubricated Stirling piston rings can be supplied with copious quantities of lubricant whereas in the internal combustion engine the lubricant fed to the top ring has to be limited to the minimum quantity adequate to prevent oil film breakdown. Surplus oil passes onto the combustion space and is burnt, thus increasing the lubricant oil consumption.

The rates of pressure increase experienced during combustion in a diesel are orders of magnitude greater than the maximum rates of pressure change in a Stirling engine where the pressure-time diagram is nearly sinusoidal. Therefore, the very severe connecting rod and crank bearing loads experienced during combustion in a diesel engine simply do not exist in a Stirling engine.

The operating characteristics of a sliding bearing are frequently expressed in terms of the bearing-characteristic or Sommerfeld Number S defined as

$$S = (r/c)^2 \mu N/P$$

where

S = Sommerfeld Bearing Characteristic No.,
r = journal radius,
c = radial clearance,
μ = absolute viscosity,
N = relative speed of the journal and bearing,
P = load per unit of projected area.

For a given bearing operating at a given speed the ratio of Sommerfeld numbers for oil and water would be:

$$\frac{S_o}{S_w} = \frac{\mu_o}{\mu_w} \times \frac{P_w}{P_o}$$

At a temperature of 125°F the absolute viscosities of oil and water are: (Shigley 1977)

1. SAE 30 oil - 5.75×10^{-6} reyn.
2. Water - 7.5×10^{-8} reyn.

Therefore:

$$\frac{S_o}{S_w} = \frac{5.75 \times 10^{-6}}{7.6 \times 10^{-8}} \frac{P_w}{P_o} = 76\ P_w/P_o$$

Clearly water is much less advantageous than oil as a lubricant. However, the high impact type loadings in the oil lubricated diesel engine during combustion compared with the water lubricated Stirling engine are thought to make the ratio of bearing loadings P_w/P_o small enough to completley eliminate the viscosity advantage of the oil. In fact the Sommerfeld Number ratio S_o/S_w is likely to be something less than unity suggesting that water would be perfectly adequate as the lubricant and coolant in an air-Stirling engine. Moreover it must be recalled that the critical piston ring lubrication problem of internal combustion engines simply does not exist in Stirling engines.

Problems of corrosion are foreseen in the use of water but these could be overcome by internal epoxy coatings on exposed steel parts where corrosion was thought to be important, the use of plastics and plated or chromed surfaces elsewhere and, for many parts, simply accepting the corrosion and making allowance in design for it. Polytetrafluoroethylene (P.T.F.E.) plastics filled with chopped glass or metal powders would likely be used for the main crankshaft, and connecting rod bearings and piston rings.

Another problem, particularly in the cold climate of Canada, is that water freezes at 0°C. The use of ethylene glycol-water mixtures for the lubricant would be permissible but could not be tolerated if the lubricant water were also used for the water-injection power boost described above. The ethylene-glycol would decompose at high temperatures and contaminate the single tube boiler, and steam injector on the hot side of the working space. No solution to the freezing process comes to mind except the obvious one of keeping the engine warm or prewarming it prior to starting.

6.5 Convert Diesel Engines to Stirling Engines

In the preceding chapter we sought to establish that coal-fired Stirling engines could be made of such size as to permit the installation in the locomotive of a Stirling engine as a direct replacement of the diesel unit. This is very attractive to permit continued use of the locomotive for the useful life remaining. Here we go further to suggest the diesel engine removed *be converted to operate as a Stirling engine* with all the features discussed above of water lubrication, air working fluid, and steam injection. Such conversions appear to be perfectly feasible

as we shall show shortly. If the process of engine conversion could indeed be achieved the savings resulting from the use of many familiar and existing parts would be staggering compared with the cost of building completely new engines for all existing locomotives as well as replacement vehicles for the future.

We shall adopt for conversion the familiar diesel engine used for the reference engine above, i.e. the MLW Series 251-C shown in Figure 20. To begin the conversion we strip the engine to the skeletal arrangement shown in Figure 29. We retain the crankshaft, crankcase, cylinder block, cylinder liners, pistons and connecting rods but dispense with all the other parts contained within the broken line *silouhette*.

6.5.1 Two-Piston Machines

One possibility that immediately suggests itself is to make the engine a Vee-form, two-piston machine as shown in Figure 30, with one cylinder as the cold compression cylinder and the other cylinder as the hot expansion space. A Heylandt crown is added to the hot piston to isolate the hot expansion space from the area where the piston seals operate on the water cooled walls of the wet liner.

The particular design shown in Figure 30 is unacceptable for the angle between the cylinder axes of the Series 251 diesel is only 45°. This means the phase angle by which volume variations in the hot space lead those in cold space would also be this angle 45°. A phase angle of about 90° is required for maximum power and angles in the range 60 to 120° are acceptable but a phase angle of 45° is simply too remote from the optimum.

Multiple cylinder engines with the cranks disposed at 90 or 120 degrees offer good possibilities to overcome this problem of an inappropriate phase angle. Some of the various ways in which the cylinders might be coupled are illustrated in Figures 31 through 35.

Consider first the six-cylinder in-line arrangement with the cranks at 120° as shown in Figure 31. According to manufacturer's literature the firing order for the diesel engine is 1-4-2-6-3-5. For the proposed two-piston Stirling conversion, cylinders 1, 4 and 6 are the hot cylinders coupled to cylinders 3, 2 and 5, respectively. Three separate, single-acting Stirling engine systems are thus created.

No satisfactory arrangement could be devised for the 8-cylinder, Vee engine, for the cranks are thought to be disposed at 180°. Reassembly of the crankshaft with 90 crank angles would permit similar couplings to those given below for the 16-cylinder (twin 8's) engine.

The 12-cylinder, Vee engine lends itself to the connections shown in Figure 32, basically twin combinations of six-cylinder modules with phase angles of (120-45) = 75°. There are six, separate, single-acting, two-piston, Stirling engine systems. All the six hot cylinders are conveniently arranged on one (right hand) side of the engine and the six cold cylinders on the left hand side.

A similar arrangement can be used for the 18-cylinder version as shown

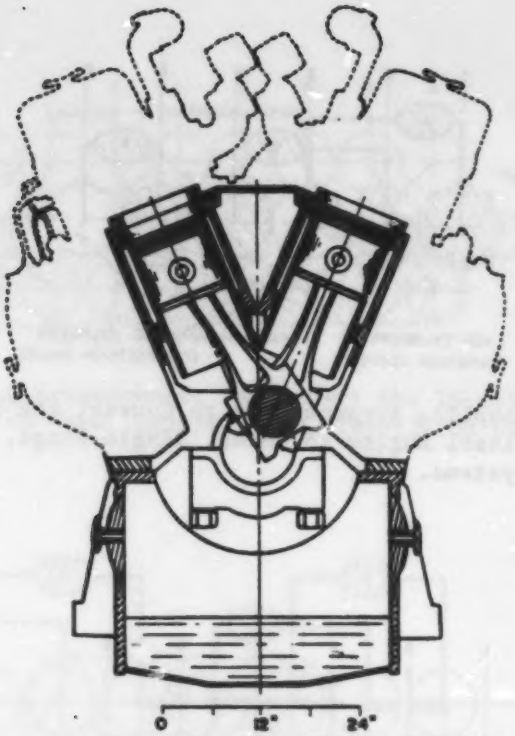


Figure 29 Skeletal Arrangement of MLW Series 251-C Diesel Engine Used as the Foundation for a Stirling Engine Conversion.

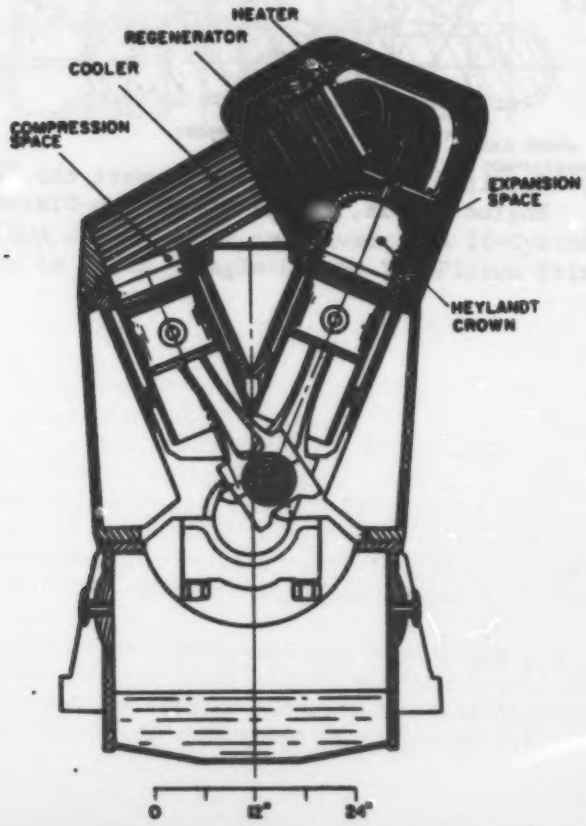


Figure 30 Vee Configuration Two-Piston Stirling Engine.

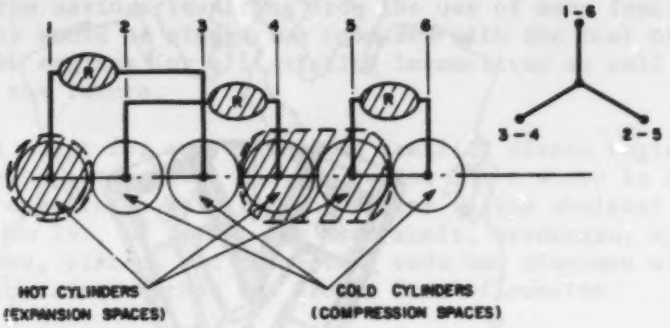


Figure 31 Coupling Arrangements to Convert the Six-Cylinder In-Line Diesel Engine to Three, Single-Stage, Two-Piston Stirling Systems.

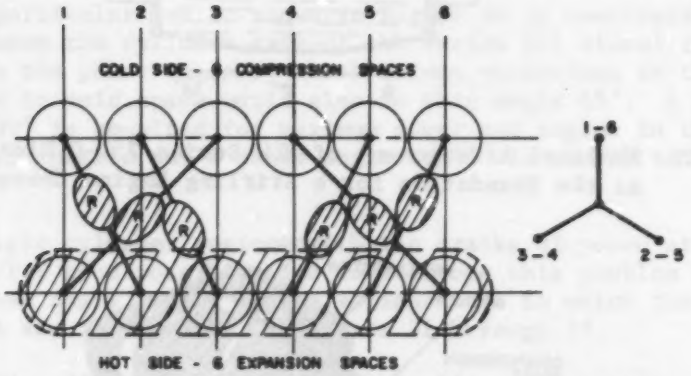


Figure 32 Coupling Arrangements to Convert the 12-Cylinder Vee Diesel Engine to Six, Single-Stage, Two-Piston Stirling Systems.

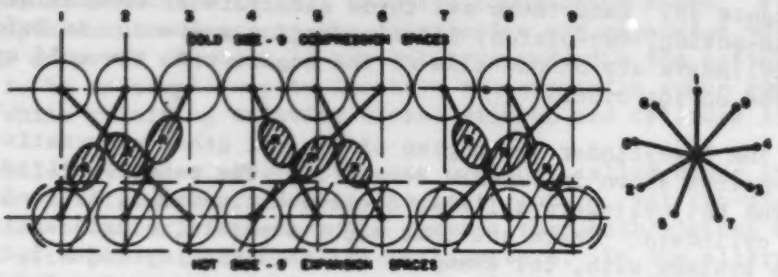


Figure 33 Coupling Arrangements to Convert the 18-Cylinder Vee Diesel Engine to Nine, Single-Stage, Two-Piston Stirling Systems.

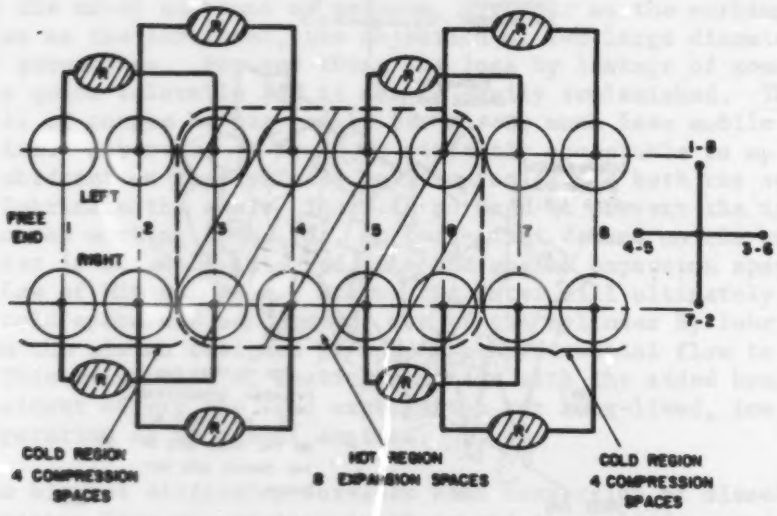


Figure 34 Coupling Arrangements to Convert the 16-Cylinder Vee Diesel Engine to Eight, Single-Stage, Two-Piston Stirling Systems.

in Figure 33. Here there are three six-cylinder modules arranged as 9 single-acting, two-piston, Stirling engine systems. As before all the hot cylinders are on one side of the engine with the cold cylinders grouped on the other.

The 16-cylinder Vee engine offers yet other alternatives with the connections shown in Figures 34 and 35. The central cylinders 3-4-5-6 are the hot cylinders while at the two extremities, 1-2 and 7-8 are the cold cylinders. There are thus eight separate, single-acting, Stirling cycle systems with, for example, the left hand, cylinder 1, compression space coupled, through a duct with cooler, regenerator and heater to the left hand cylinder 3, expansion space. Similarly, a right hand side

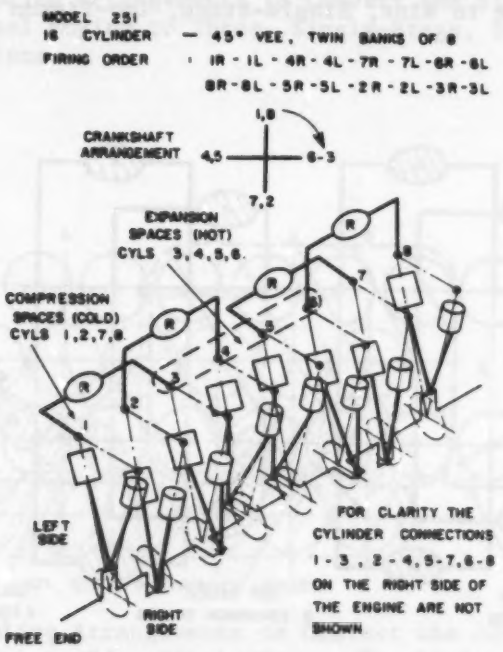

MODEL 251
16 CYLINDER — 45° VEE, TWIN BANKS OF 8
FIRING ORDER : 1R - 1L - 4R - 4L - 7R - 7L - 6R - 6L
8R - 8L - 5R - 5L - 2R - 2L - 3R - 3L
CRANKSHAFT ARRANGEMENT
1,8
4,5
6-3
7,2
EXPANSION SPACES (HOT) CYLS 3,4,5,6.
COMPRESSION SPACES (COLD) CYLS 1,2,7,8.
LEFT SIDE
RIGHT SIDE
FREE END
FOR CLARITY THE CYLINDER CONNECTIONS 1-3, 2-4, 5-7, 6-8 ON THE RIGHT SIDE OF THE ENGINE ARE NOT SHOWN

Figure 35 Coupling Arrangements for the 16-Cylinder Vee Engine to Four or Eight, Single-Stage, Two-Piston Stirling Systems.

cylinder 1 is coupled to right side cylinder 3.

An alternative and more attractive arrangement would be to combine the left and right hand, number 1 bank, cylinders to a single compression space and the left and right hand number 3 bank cylinders to a single expansion space. These combined expansion and compression spaces would then be coupled by a single heater, regenerator and cooler combination instead of two. The complete engine would thus consist of *four* separate single-stage Stirling systems, a more attractive combination than the *eight* separate systems described earlier.

Stirling engines of the two-piston type, converted as described above from diesel engines,, would have to have the same swept volumes in all cylinders since the bore and stroke is the same for all the diesel cylinders. Now for optimum operation of a Stirling engine with expansion

space temperatures of 800°C, the expansion space swept volume should be larger (about one-third larger) than the compression space. With equal displacement volumes in the compression and expansion volumes there is a loss of power of about 10 percent compared with the optimum. There appears to be no simple way to increase the expansion space swept volumes while retaining existing diesel pistons and cylinder liners.

Parallel cylinder, two-piston air engines, called Rider air engines were used in large numbers during the last century, yet two-piston Stirling engines are often criticised on the grounds that both pistons must be provided with pressure seals of large diameters. In the alternative piston-displacer machines the pressure seals for the displacer rod are much smaller than the piston seal and therefore consume less frictional power and provide a smaller leakage path. This is a prime advantage of the piston-displacer arrangement. Of course a large diameter seal is required on the displacer but this is less critical than the *hermetic* seal required for the piston.

In the novel machines we propose, with air as the working fluid and water as the lubricant, the objection of two large diameter seals becomes groundless. For one thing the loss by leakage of some air working fluid is quite tolerable for it can be easily replenished. The leakage rate will of course be very small for air is much less mobile than hydrogen and helium. Moreover, it would be perfectly acceptable to splash excess water lubricant on the cylinder wall to facilitate both the sealing function and to lubricate the seal. There is no need to prevent the ingress of water to the working space for, at peak-power demand on the engine, the same water is actually injected into the engine expansion space. Because of the Law of Minimum Energy States the water will ultimately accumulate in the cold space and be 'pumped' out of the cylinder by lubricant scraper rings on the piston designed to induce a preferential flow to the crankcase. This relaxation of sealing function with the added bonus of plentiful lubricant offers the same expectation for long-lived, low friction power operation as in diesel engines.

The biggest difficulty foreseen with conversion of diesel engines to two-piston Stirling engines is that half the cylinders are given over to operate as compression spaces and half as expansion spaces. In Section 6.2 data was presented for the power densities of Stirling engines and diesel engines in terms of brake horsepower per litre piston displacement. This data refers specifically to the *piston* displacement of a piston-displacer type Stirling engine. Now if, in the two-piston machine, we allocate only half the total piston displacement to compression space then, clearly, we have only half the total engine displacement to allocate for the power output per unit displacement shown in Figure 26. In other words the two-piston Stirling air engine would have only *half* the power output of the diesel engine. Of course it would be possible to increase engine output by raising the pressure and speed and by power boost steam injection but there is insufficient data to permit a reliable prediction about thermal efficiency although a substantial decrease might be anticipated.

6.5.2 Ringbom Engines

To maintain both high engine output and efficiency we must therefore use the total displacement of all pistons as compression space and add the necessary displacer cylinders to provide expansion space. It would be possible to drive the displacers by a kinematic linkage connected to the crankshaft. Some use could likely be made of the camshafts for this purpose but it would be complicated and fortunately is not necessary. A simpler and better way is to use the Ringbom Stirling engine shown in Figure 36. Here a conventional internal combustion engine cylinder with piston, crank, connecting rod and crankcase is coupled to another cylinder containing the displacer and shown immersed in a fluidised bed with steam injection to the expansion space. The motion of the displacer arises from fluidic forces imposed on it by the variation in the working space pressure and the pressure of the displacer gas spring. Space precludes a full discussion of the mechanics and dynamics of free-piston Stirling engines but a complete treatment is given in Walker *et al* (1982). It is sufficient to say here that the motion of the piston and displacer and the consequent volume and pressure variations in the compression, expansion and spring spaces are approximately as shown in Figure 37.

The displacer may oscillate in the resonant mode wherein the motion

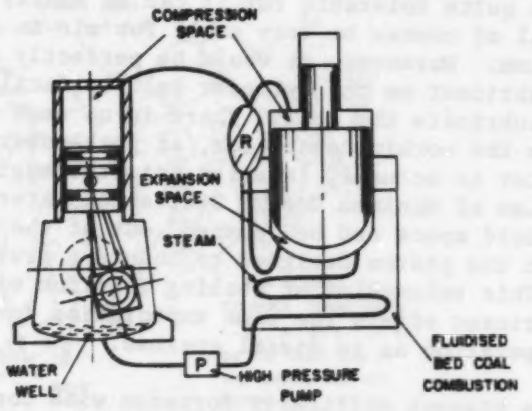


Figure 36 A Ringbom-Stirling Engine wtih Fluidized Bed Heat Source and Power-Boost Water Injection.

is nearly simple harmonic as shown in Figure 37 or may be 'overdriven' with an approximate square-wave motion and long dwell periods at the top and bottom stroke positions. This is often described onomatopoetically as a 'bang-bang' engine.

The motion of the displacer is affected greatly by the 'stiffness' of the pneumatic spring. A mechanical spring could be used but would be relatively massive and has other disadvantages as well as some advantages compared with a gas spring.

Stiffness of a gas spring (the force required to compress the spring by unit deflection) is given approximately by the equation:

$$k = \frac{\gamma p A^2}{V_o}$$

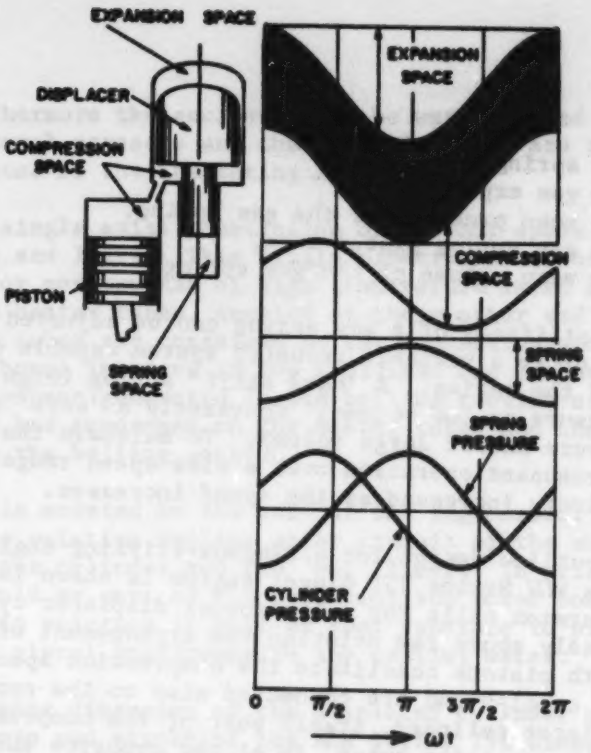


Figure 37 Piston and Displacer Motion with Consequent Volume and Pressure Variations in the Ringbom Stirling Engine.

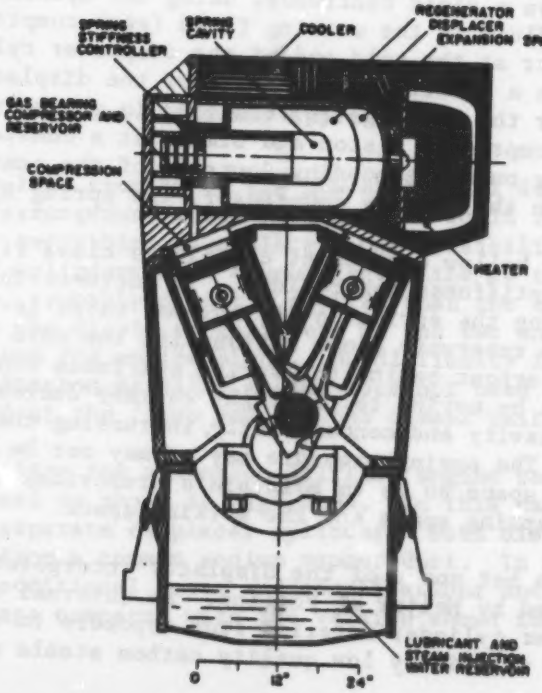


Figure 38 Sketch of Ringbom-Stirling Engine Based on the MLW Series 251 Diesel Engine Conversion with One Displacer Cylinder per Pair of Cylinders.

where

k = spring stiffness,
γ = gas exponent,
p = mean pressure of the gas spring,
A = gas spring piston area,
V_o = mean volume of the gas spring.

The stiffness of a gas spring can be adjusted by change in the mean pressure or by a variable geometry system capable of changing the mean volume of the spring. A 'hard stiff' spring (high k) is produced by a high pressure or low volume. Conversely a 'soft' spring results from a low pressure and/or large volume. To maintain the displacer in resonant or near-resonant operation over a wide speed range the stiffness must be progressively increased as the speed increases.

An outline design for a Ringbom-Stirling engine based on the conversion of a MLW Series 251 diesel engine is shown in Figure 38. Basically the conversion calls for a horizontal displacer cylinder to be mounted transversely above the existing Vee arrangement of the pistons. The space above both pistons constitute the compression space. The two spaces are mutually coupled and are connected also to the space at the left end of the displacer cylinder, itself part of the compression space. The hot space, the expansion space, is at the opposite end of the displacer cylinder.

The displacer is situated between the compression and expansion spaces. It oscillates in the cylinder causing fluid to move through the cooler, regenerator and heater to the hot space and vice versa. The displacer is shown in the so-called 'centre-post' arrangement with the spring cavity contained in the displacer. The displacer is mounted coaxially on a rigid centrepost using two hydrostatic gas bearings. The bearing fluid is the working fluid (air) compressed in a bearing gas compressor at the cold end of the displacer cylinder. The compressor piston is a concentric extension of the displacer. The compressed working fluid for the bearings is contained in a reservoir in the displacer or the compressor piston and bleeds at a constant rate to the gas bearings operating on corresponding journals of the centrepost. The bearing fluid drains to the bearing compressor, the spring space or the working space.

The hollow centrepost contains a close fitting piston called the 'spring stiffness controller'. To increase the spring stiffness, by decreasing the spring cavity volume, water is driven pneumatically, from the reservoir, shown surrounding the cold end of the displacer, into the centerpost behind the spring stiffness controller piston. The piston responds by moving to the right thereby decreasing the volume of the spring cavity and, concomitantly, increasing the mean pressure in the spring space. The spring pressure may or may not be throttle-vented to the working space so as to maintain a prescribed mean pressure relationship of the spring space and the working space.

The hot space of the displacer incorporates other design features pioneered by Meijer at Philips. An internal ceramic liner protects the displacer cylinder material from exposure to the hottest gas temperatures so that relatively low quality carbon steels may be used for the cylinder

enclosure. Furthermore the enclosure may be water-cooled to further diminish the thermal stresses and the thermal losses are relatively insignificant because of the insulating liner.

There is a single axial penetration of the hot space enclosure for the gas to enter and leave. This single penetration is coupled to a header/distributor constructed of high temperature rated alloy, and connected to the heater tubes, coupled at their other end to the regenerator. The heater tubes are contained in the condensing section of a sodium heat pipe (not shown) immersed in the fluidized bed combustor (not shown). Sodium vapour generated in the boiling section of the heat pipe in the fluidized bed condenses on the heater tubes and the liquid returns under gravity to the boiling section.

The cooler is mounted to the left of the regenerator and would be integrated in the existing cooling water circuit of the engine through the annulus between the cylinder and the 'wet' liner. In principle, the cooling water could be part of the lubricant and power boost water used internally, but in practice it will be kept separate to permit the inclusion of ethylene glycol antifreeze in the cooling water.

The approximate dimension of the displacer cylinder may be determined from the known bore and stroke of the MLW Series 251 diesel engine, 9 in. and 10.5 in., respectively. The combined swept volume V_C of the left and right hand pistons are $2 \times 668 = 1336$ in^3. Since the optimum ratio $V_C/V_E = 0.75$ then the swept volume V_E of the expansion space is $V_E = 1336/0.75 = 1781$ in^3. Assuming the displacer stroke L to be half the displacer diameter D then we have:

$$V_E = \frac{\pi}{4} D^2 L = 1781 \text{ in}^3$$

or

$$D = \sqrt[3]{\frac{1781 \times 8}{\pi}} = 16.5 \text{ ins. (419 mm) approx.}$$

The cylinder centre-line spacing on the MLW Series 251 diesel is about 18.5 ins. so the design arrangement shown in Figure 28 would require careful manipulation to get everything into place. One possibility would be to place the displacer cylinders on a common centreline at the minimum possible centre distance, probably a little greater than the existing cylinder centre distance, so the displacer cylinders at the two ends would likely extend some way beyond the engine block. No difficulty is anticipated about this for the Ringbom-Stirling engine does not need the large turbocharger mounted at the 'free end' of the diesel unit.

Another possibility for a Ringbom-Stirling engine based on the same Series 251 diesel is shown in Figure 39. In this case each piston is provided with a separate displacer cylinder. Both displacer cylinder heaters are served from a common sodium vapour duct. In this arrangement there appear to be additional costs and complications and little compensating advantage compared with the version shown in Figure 38.

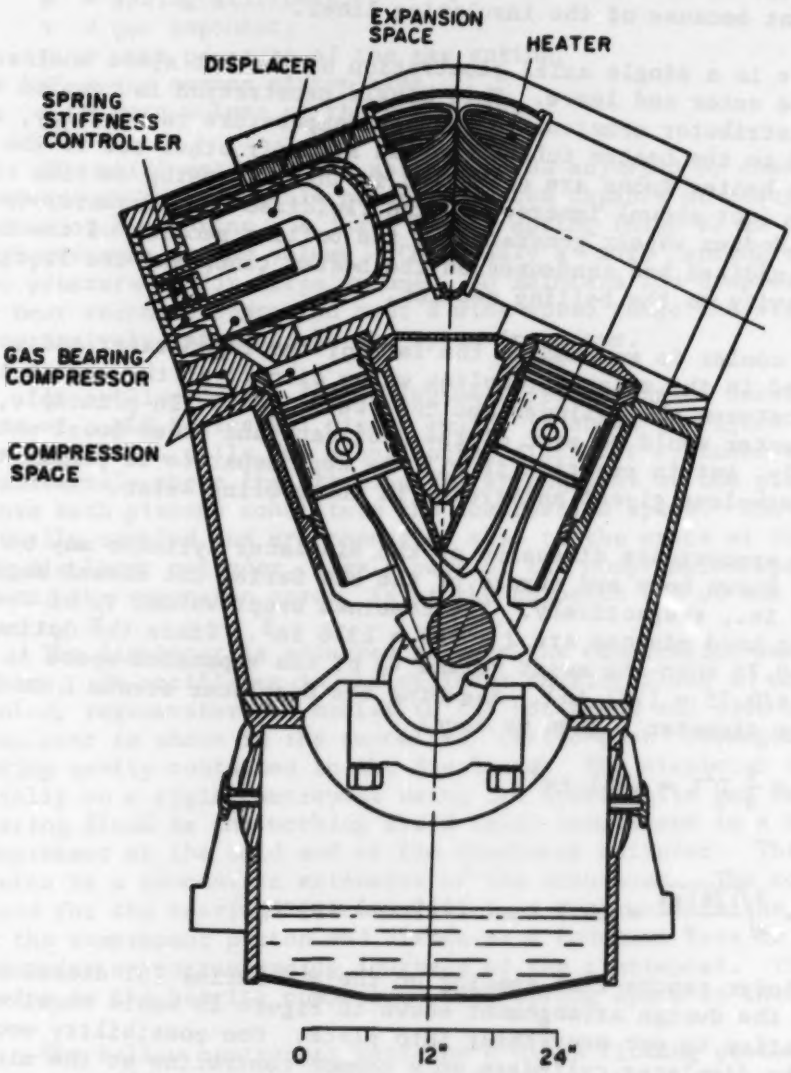


Figure 39 Sketch of Ringbom-Stirling Engine Based on the MLW Series 251 Diesel Engine Conversion with Individual Displacer Cylinders per Cylinder.

Yet another alternative arrangement is shown in Figure 40. Here one bank of pistons has been sacrificed to permit a reduced installation envelope but, of course, at the expense of half the engine power.

The bore/stroke ratio, (9/10.5) of the MLW Series 251 diesel is not the most favourable ratio for Stirling operation. For the reasons discussed by Walker (1980) the optimum bore/stroke ratio for Stirling engines is about two to one. Superficial inspection suggests the cranks, bearings and other mechanical parts of the Series 251 diesel are generously proportioned and could likely sustain much higher stresses in Stirling engine operation where the loadings are more or less steady and without the violent, impact-like, variation of the combustion process in diesel engines. With this in mind, a Ringbom Stirling engine of more favourable bore/stroke ratio has been sketched in Figure 41. The Series 251 stroke of 10.5 ins. has been retained but the bore has been increased to 18 ins. The piston displacement volume is then:

$$V_C = \frac{\pi}{4} \times 18^2 \times 10.5 = 2670 \text{ in}^3,$$ i.e. four times that of the conventional piston.

This 'high power' version of the engine could be a development project for new locomotive equipment following a period of successful conversion of existing diesel fleets to Ringbom-Stirling air-and-water-engines.

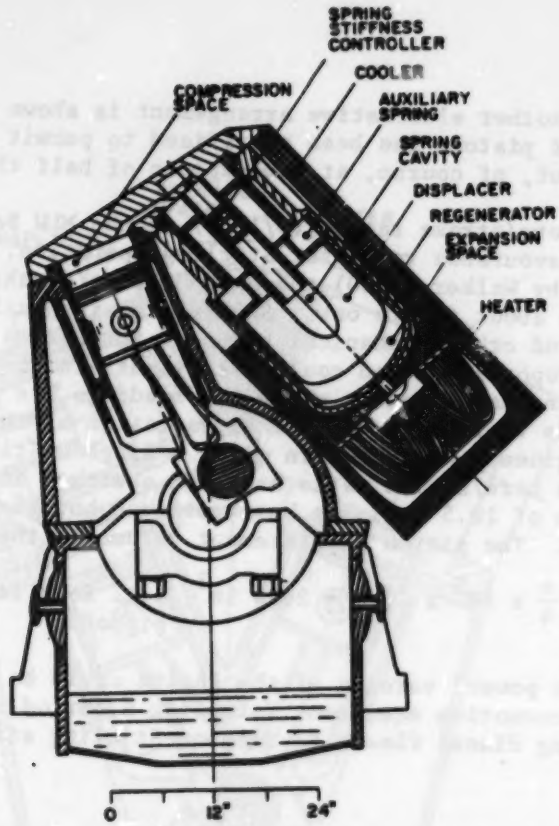


Figure 40 Sketch of Ringbom-Stirling Engine wtih One Bank of Cylinders Replaced by the Displacer Cylinders.

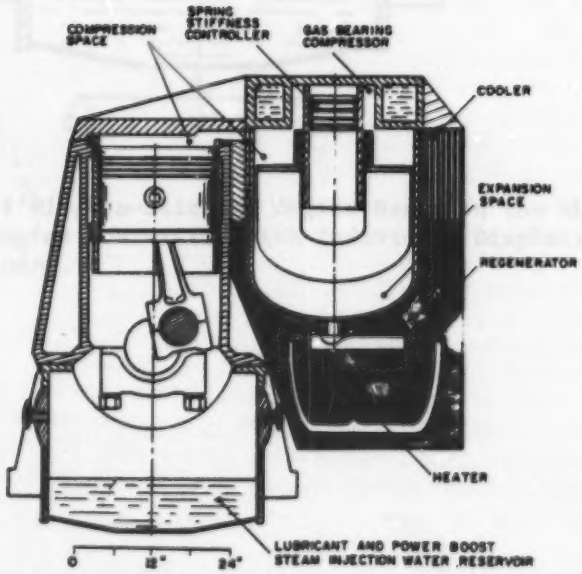


Figure 41 Ringbom-Stirling Engine with a Large Bore/Stroke Ratio for the Piston.

7. THE COAL-BURNING STIRLING ENGINE LOCOMOTIVE

The Stirling engine locomotive will in external appearance likely resemble contemporary diesel units. Figure 42 is a conceptual view of a coal-burning Ringbom-Stirling/electric powered locomotive modelled after the Bombardier MLW M630 six-axle, 3000 horsepower diesel-electric locomotive.

The principal features of interest are:

a) use of a Ringbom-Stirling engine as the propulsion unit installed in the same location as the conventional diesel unit.

b) fluidized bed coal combustor located beneath the locomotive frame between trucks (bogies) in the space normally containing the fuel tanks. The fluidized bed will likely occupy half to three quarters the available space leaving room for a small buffer. The locomotive is shown equipped with twin fluidized beds capable of independent operation and energizing half the engine. For cruise or idle conditions with low power requirement one fluidized bed would be turned down to the minimum necessary to sustain the bed in a state of active readiness. Independent operation of the beds would likely be incorporated in the locomotive control system coupled to the throttle with a manual over-ride.

   Twinning the fluidized beds has the further advantage that even with one of the beds inoperable, due to some system failure, the engine could still produce half-power.

c) a sodium heat pipe heat transport system coupling the fluidized bed combustor to the engine list spaces. Four heat pipes are shown, each feeding two displacer cylinder heater heads. In practice it is most likely each of the four units would actually contain a cluster of vapour feed pipes with perhaps a single liquid downcomer. The whole cluster would be contained in a heavily insulated duct as shown.

d) long range operation requires the provision of a coal tender attached to the locomotive containing prepared coal (1/4 in. (6 mm) diameter screened). The coal would be conveyed by an electrically driven Archimedean screw auger from the tender to the buffer store and thence to the fluidized beds as required. In cold climates the warm exhaust gases leaving the exhaust gas/air preheater could be conveyed through the coal tender to preclude freezing and consolidation of the coal particles.

e) additional engine cooling water air-cooled radiator surface would be incorporated by enlarging or double-banking the existing radiators and by adding extra radiator units. Stirling engines require about twice the radiator cooling surface of diesel units. In a Stirling engine the heat dissipated directly to exhaust must be minimized (to 5-10 percent of the heat input) rather than one-third the heat supplied of a water

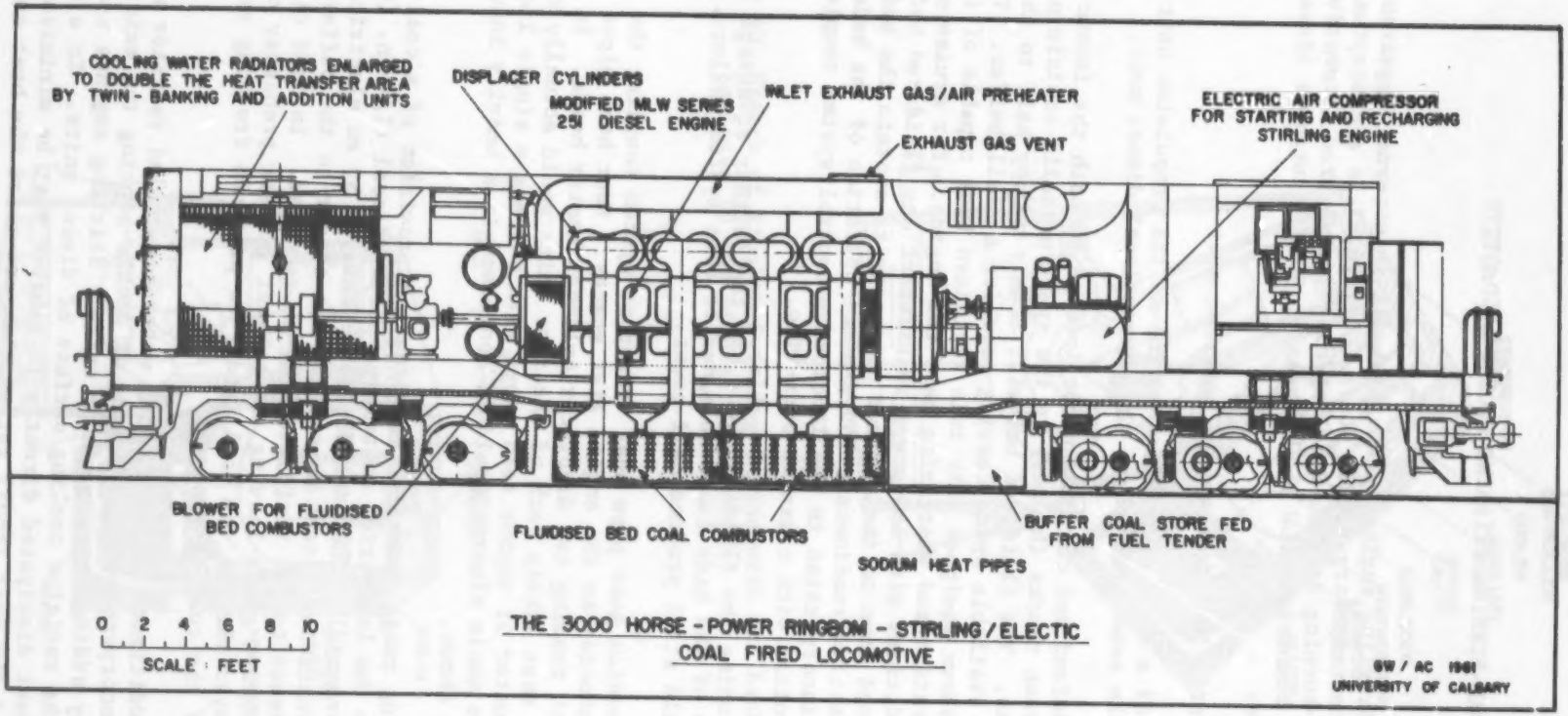

COOLING WATER RADIATORS ENLARGED TO DOUBLE THE HEAT TRANSFER AREA BY TWIN - BANKING AND ADDITION UNITS
DISPLACER CYLINDERS
MODIFIED MLW SERIES 251 DIESEL ENGINE
INLET EXHAUST GAS / AIR PREHEATER
EXHAUST GAS VENT
ELECTRIC AIR COMPRESSOR FOR STARTING AND RECHARGING STIRLING ENGINE
BLOWER FOR FLUIDISED BED COMBUSTORS
FLUIDISED BED COAL COMBUSTORS
SODIUM HEAT PIPES
BUFFER COAL STORE FED FROM FUEL TENDER
0 2 4 6 8 10
SCALE : FEET
THE 3000 HORSE - POWER RINGBOM - STIRLING / ELECTIC COAL FIRED LOCOMOTIVE
SW / AC 1981
UNIVERSITY OF CALGARY

Figure 42 The Stirling Engine Locomotive.

cooled diesel unit. The balance of heat rejected from the Stirling engine must therefore pass through the engine cooling system. Furthermore a high cooling temperature has a serious deleterious effect on Stirling engine efficiency, whereas it actually improves the efficiency of a diesel engine if not carried so far as to cause lubrication breakdown. Cooling the cooling water for Canadian operations does not have the same challenge as operations in hot climes.

f) an electrically driven air compressor and reservoir are provided to supply compressed air for starting and to recharge the Stirling engine or replenish leakage. Starting air would be required to initially fluidize the bed and maintain it fluidized during the warm-up starting period with heating of the bed by propane or natural gas. With the bed heated the engine could be started using the normal starter motor of diesel/AC electric systems. Another possibility would be to use the compressed air to move the spring stiffness controller pistons in the displacer cylinders. This would cause the displacers to move in response to the gas spring changes. Motion of the displacers causes pressure variations in the working space sufficient to start the engine. Even single-cylinder Ringbom-Stirling engines are virtually self-starting and a multiple-cylinder version would require very little effort to start once the heater head was warmed beyond the minimum starting temperature.

g) a large, low pressure fan driven by the engine would provide air for the fluidized bed combustors.

h) a very substantial exhaust gas/inlet air preheater would be located above the engine in the space conventionally taken up by the diesel muffler. The inlet air preheater would reduce the coal consumption by abstracting heat from the fluidized bed effluent before release through the exhaust gas vent.

i) an electrically or engine driven, high pressure water pump, not shown on Figure 42, would feed liquid water in appropriate quantities to the once-through, single tube boiler generating steam for the power-boost steam injection system used for peak power production.

j) everything else on the locomotive would be more or less the same as on the existing diesel powered version. The electric power generator and controls, the traction motors, the locomotive structure, the frame, trucks, wheels, etc., cooling fans would all remain identical. The diesel engine muffler would be removed to produce space for an exhaust gas/inlet air preheater or, more likely, the additional air-cooled radiator surface necessary with Stirling engines.

8. OPERATING CHARACTERISTICS OF THE STIRLING ENGINE LOCOMOTIVE

8.1 Low Noise

The most noticeable difference in the operating characteristics of Stirling and diesel engines would be the low noise level. Stirling engines run silently for there are no valves or periodic explosions in the cylinder as in the diesel engine. Usually the principal noise in a Stirling engine arises from the fans used for moving the cooling air. This low noise level is, of course, the principal environmental advantage of a Stirling engine locomotive and one that would be welcomed universally by Canadians nightly exposed to the intense nocturnal activity favoured by railroad operators.

8.2 Starting

Starting the Stirling locomotive would be less rapid and convenient than a diesel. The fluidized bed must be 'warmed through' by combustion of propane or natural gas before it can be properly fluidized and made self-sustaining. The procedure might be expected to take 15 to 30 minutes and require the provision of gas and compressed air. These might be contained on board or it might become standard practice to start the locomotive only at established bases.

Starting the engine after warm-up could be accomplished using the conventional diesel starter motor or by using compressed air to move the spring stiffness control pistons in the displacer cylinders to cause a pressure perturbation and the engine to self-start.

8.3 Load Response

The coal-burning fluidized bed would be substantially less responsive to sudden load changes than the virtually instantaneous response obtained from a diesel engine. However, the Stirling engine operating parameters can be adjusted to provide a rapid response to a sudden load change while adjustments to the coal supply and the fluidized bed operating conditions take care of the longer term (5-10 min.) situation.

8.4 Environmental Pollution

The fluidized bed combustor is able to consume coal with minimal atmospheric contamination. The relatively low temperature of operation precludes the formation of NO and the addition of limestone and other agents prevent the formation of SO_2 and other environmental pollutants. Nevertheless the combustion of coal does result in the formation of appreciable quantities of ash. In a properly driven fluidized bed the ash will be in the form of fine dry powder rather than the hard fused 'clinker' common with 'stoker-grate' systems.

8.5 Ash Disposal

Disposal of powder ash from the fluidized bed combustor would be simply accomplished by continuously dumping it on the road between the rails. This may be deemed environmentally unacceptable in which case the

alternative would be to provide an 'ash-box' probably in the fuel tender for progressive accumulation of the ash and periodic disposal at designated bases. Pneumatic transport of the ash, perhaps by the exhaust gases, is a possibility with a cyclone separator in the fuel tender above the ash-box.

8.6 Coal Consumption

Kneiling (1977) suggests conventional diesel electric units, on long runs with a fuel tank car attached, consume 2 (U.S.) gallons of diesel oil per 1000 gross ton miles. Thus two power units hauling a 5000 ton train use 10 (U.S.) gallons per loaded ton mile. A 1000 mile haul and an empty return trip consume 15000 (U.S.) gallons per round trip.

Assuming a specific gravity for the oil of 0.93 and calorific values for the oil and coal of 18700 and 12000 BTU/lb respectively (43500 and 28000 kJ/kg), we can calculate the coal-fired Stirling locomotive, having the same efficiency as the oil-fired diesel locomotive, will consume 23 lbs. of coal per 1000 gross ton-miles compared with the diesels 2 (U.S.) gallons of oil. The two power unit Stirling train on a 1000 mile haul loaded and empty return trip would consume 77 tons of coal compared with 15000 U.S. gallons of oil for the diesel units.

8.7 Regenerative Braking

A diesel/electric locomotive has the facility that the DC drive motors can be rapidly switched to function as power generators absorbing energy from the train to generate DC electric power. This is useful to decellerate the train when stopping or to absorb potential energy when descending long grades. Unfortunately there is no way to use the generated power and it is simply dissipated in resistance banks.

With a Stirling engine locomotive it would be possible to store the electrical power in a thermal storage 'battery' and then reuse the power to drive the engine when required, a truly regenerative operation.

The combination of Stirling engine and thermal battery for automotive application has been extensively explored. Meijer (1970) presented the earliest data and more recently Morgan (1978) has given a very comprehensive report on thermal storage/Stirling engine systems. Lithium fluoride is often favoured as the heat storage medium because of its high latent heat of fusion at an appropriate temperature for Stirling engine operation.

8.8 Engine Overspeed

Experience has shown that Ringbom-Stirling engines operate over a limited speed range which depends primarily on the stiffness of the displacer gas spring and the applied load. This inherent limitation of engine maximum speed prevents engine runaway in the event of accidental failure of the electrical generator constituting the engine load. The control system would automatically reduce fuel flow to the fluidized bed and decrease the displacer spring stiffness.

9. THE DIRECT AND HYDROSTATIC DRIVE STIRLING LOCOMOTIVE

In the preceding section we considered locomotives in which the diesel propulsion unit was simply replaced by a Stirling engine of equivalent form and power. This was done in the light of the enormous investment and very large numbers of existing diesel locomotives. The possibility of simply replacing the power unit while extending the life of the remainder of the system has obvious and compelling advantages.

For new replacement equipment the same arrangement might be followed but alternative schemes appear to warrant further investigation including the direct drive Stirling engine locomotive and the combination of Stirling engine and hydrostatic drive.

9.1 Direct Drive

The reciprocating engines of steam locomotives are directly coupled to the driving wheels. This is possible because the torque of a reciprocating steam engine is greatest at zero speed and declines progressively with increase in speed. With the variable cut-off feature incorporated in the valve gear controlling steam admission the declining torque/speed characteristic can be greatly enhanced. Very high starting torques can be deployed with direct coupling to the drive wheels. The diesel engine does not have this advantageous torque/speed curve so a relatively complicated mechanical, hydraulic or electric transmission system is required to gain the torque amplification necessary for starting.

The conventional piston-displacer Stirling engine has a relatively flat torque/speed curve, somewhat more advantageous than the diesel characteristic but much less suited than the steam engine for direct coupling to the roadwheels. There is some prospect however, that Ringbom-Stirling engines could be endowed with the same advantageous torque speed characteristic of the steam engine. It is well known that free-piston Stirling engines are very difficult to stall, or to prevent from self-starting. Attempts to minimize the piston motion or reduce the piston stroke simply increases the internal gas forces acting to move the piston. It is virtually impossible to prevent all movement of the piston. It is quite possible the hybrid Ringbom engine will be found to possess the equivalent characteristic of increased torque at lower speeds with increased resistance to rotation.

If this were found to be the case the effect could be enhanced by increasing the charge pressure in the engine to the maximum permissible on strength considerations. Remember that high density (resulting from high pressure) is only disadvantageous at the maximum power/maximum speed condition. Very high charge pressure would be very advantageous at the starting, zero speed condition.

Limited experience with model single cylinder Ringbom engines indicates they are self-starting in the same way as free-piston Stirling engines. If this proves not to be so for the big Ringbom-Stirling

engines discussed above a guaranteed start could be achieved using the displacer cylinder spring stiffness controller. Movement of the controller piston in the centrepost cylinder would cause the displacer spring pressure to change and so move the displacer causing a pressure change on the drive piston. Cyclic admission and release of water to the spring stiffness controller cylinder would result in cyclic increase and decrease in the displacer spring characteristic with concomitant movement of the displacer resulting in variable drive forces on the piston. Very low frequency stroking of the displacer is frequently observed as the precursor to a self-start with a free-piston Stirling engine.

This inherent self-starting capability, perhaps triggered by the spring-stiffness controller, when combined with very high charge pressures for starting would permit the consequent torque/speed characteristic of the engine to be tailored to suit the requirements. It would correspond almost exactly with the variable cut-off characteristic of steam engines and direct coupling to the road wheels would appear to be entirely feasible.

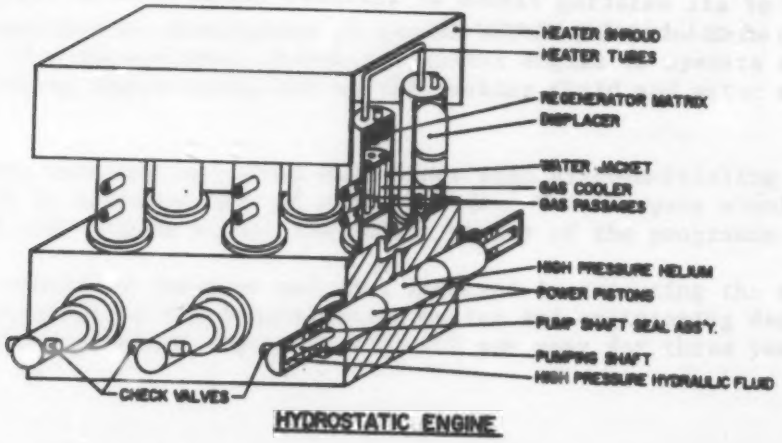


Figure 43 Orthogonal Cylinder Free-Piston Stirling Engine for Hydrostatic Vehicle Application, (after Beale 1975).

9.2 Hydrostatic Drive

Free-piston Stirling engines are best suited for operating high pressure fluid pumps utilizing linear motion directly. One concept for a hydrostatic vehicle drive utilizing free-piston Stirling engines to generate the pressurized fluid was discussed by Beale (1975). An impression of one form of orthogonal cylinder opposed piston Stirling engine pump unit for such a system is shown in Figure 43. There is every possibility this arrangement could be engineered for a Stirling engine locomotive drive system. Hydrostatic drive systems for locomotives are favoured by some European manufacturers using diesel driven pumps. This is no reason to doubt that coal-fired free-piston Stirling engine pump units could be developed to replace the existing diesel driven pumps.

10. CONCLUSIONS

Given the very conservative rating of automotive diesel engines it appears possible to devise a coal-fired Stirling engine to replace diesel engines in railroad locomotives. The Stirling engine would have about the same efficiency, and flat part-load characteristic as the diesel engine and would produce the same power within the same installation envelope. The use of water as the engine lubricant and air as the working fluid would overcome existing problems with seals in the high specific output Stirling engines for automotive use. Power boost with steam injection would provide the air-Stirling engine the possibility of gaining the same power output as a diesel engine for the relatively brief periods of operation at maximum power or conventional locomotive operation.

It appears possible to convert existing diesel engines to operate as Ringbom-Stirling engines with no loss of power, efficiency or with any increase in size or weight. This would allow the gradual conversion of all existing fleets of railroad locomotives to coal-burning instead of oil-burning operation.

11. RECOMMENDATIONS

This brief study has shown the Stirling engine has great potential for application to railroad locomotives given the need to use coal instead of oil as the fuel. Further work along the same lines of the present study appears justified.

A more extensive engineering design should be prepared for the Ringbom-Stirling air engine described herein. The design should include detailed consideration of the sodium heat pipe heat transport system, the heater head design, the fluidized bed system and the conversion of the diesel engine to a Ringbom-Stirling engine.

Final design would not be advisable until a small proof-of-principle engine has been satisfactorily demonstrated. It is therefore recommended that experimental development of a 20 horsepower demonstration engine be undertaken in concert with, or even preceding, the engineering design outlined above.

The experimental development programme should take the form of the conversion of a two-cylinder industrial diesel engine to operate as a Ringbom-Stirling engine using air as the working fluid and water as the lubricant.

The only existing practical experience with Ringbom-Stirling engines is possessed by Sunpower Inc. of Athens, Ohio. This company should be retained as consultants in all the design phases of the programme.

It is estimated the time and cost involved in executing the experimental development of the demonstration engine and engineering design of the prototype locomotive engine is $250,000 per year for three years.

PART II - GAS-FIRED STIRLING ENGINES FOR LOCOMOTIVE PROPULSION

12. LIQUID ENERGY GASES

12.1 Introduction

Following completion of the above work further studies were commissioned concerned with the combination of gaseous fuels and Stirling engines for locomotive propulsion. Specific task assignments embraced comparison of direct-fired and indirect-fired heaterhead assemblies, the outline design of a gas fuelled Stirling engine locomotive and identification of the gas-fuel characteristics and risks involved.

For the purpose of this study the candidate, gaseous fuels were assumed to include the low order paraffin hydrocarbons butane, propane, ethane, methane and also hydrogen. All these gases, with exception of hydrogen, may be classified as primary fuels being customarily obtained by relatively minor processing of the natural gas gathered in the production of crude oil or natural gas and coal mining. Hydrogen is not a primary fuel and is not readily available in large volume without very substantial input of energy, more than can be recovered by subsequent combustion or combination in a fuel-cell with oxygen. Its use as a fuel is predicated on the availability of low cost electrical power for electrolysis to produce the fuel with the possibility for subsequent storage and transmission in gas pipelines or reservoirs with eventual use in fuel cells or combustion systems. Hydrogen is principally used at the present time as a chemical feedstock for ammonia (NH_4) production, petroleum refining methanol production plastics manufacturing, hydrogeneration of edible oils and direct reduction of iron ore and other metallurgical processes. Current demand for hydrogen is small compared with hydrocarbon fuels, about 2% on an energy basis. Industrial hydrogen is produced by steam reforming of natural gas (CH_4) or oxidation of heavy petroleum oils. There would be little purpose in utilizing either of these fuels to produce hydrogen as a fuel for use as an alternative to hydrocarbon fuels. Kukkonen (1981) has given a good review of hydrogen as an alternative automobile fuel.

12.2 Liquid Energy Gases

Butane, propane, ethane, methane and hydrogen are collectively called 'liquid energy gases' or 'cryogenic fuel gases' because they are carried and stored in the liquid state at elevated pressure or reduced temperature in thermally insulated or non-insulated containers. Propane and butane (often existing in mixtures of varying proportion called 'liquid petroleum gas', LPG) can be stored in uninsulated steel tanks at normal temperatures with internal pressures of 1.5 MPa at 310 K and lower pressures at lower temperatures.

The vapour pressure of ethane at normal atmospheric temperatures is much too high for tank storage in the liquid state at atmospheric temperature. Similarly the critical temperature of methane (190 K) and hydrogen

(200 K) is less than ambient atmospheric temperatures, so these gases simply cannot exist as liquids at normal temperatures whatever the pressure. They must therefore be cooled below their liquefaction temperature and contained in highly insulated tanks if they are to be stored in the liquid state. It is customary for 'cryogenic' storage tanks to be largely unpressurized with venting systems set to open slightly above atmospheric pressure.

The storage temperature, or boiling point at normal atmospheric pressure, of the various gases is:

Hydrogen - 20 K
Methane - 112 K
Ethane - 185 K
Propane - 231 K
Butane - 272 K

The descriptor 'cryogenic' (derived from the Greek term for icy cold) is best reserved for use in the temperature regime below 120 K. Unfortunately the term is loosely applied by many, including the transportation regulatory authorities in North America, to embrace all the customary liquid energy gases.

Table II.1 contains a summary of various properties of air and the fuel gases hydrogen, methane, propane and butane.

Very comprehensive data on gaseous fuels has been compiled by Rose and Cooper (1977). Various aspects of the flammability of gases and vapours have been summarized and discussed by Coward and Jones (1953) and by Zabetakis (1965).

A comprehensive study* of the transportation environment for the movement of 'cryogenic' fuel gases in Canada was carried out in 1979, principally by this author, for the Transport of Dangerous Goods Branch of Transport Canada, Ottawa. The study in five volumes includes various aspects of road, rail, inland waters and marine transportation of liquid energy gases including the risks inherent therein.

12.3 Combustion of Liquid Energy Gases

All the liquid energy gases combust readily in air or oxygen and release large quantities of energy. This is advantageous from the aspect of their use as fuels but also constitutes a principal hazard in their storage, transportation, use and disposition following an accident resulting in an uncontrolled release or 'spill' of the liquid gas. Unfortunately the technology does not exist to render them harmless during transportation or storage and to reactivate them for use as fuels.

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*Study to Investigate the Transportation Environment of the Movement of Cryogenic Fuel Gas and the Development of a Risk Analysis Model (in 5 volumes) (1979), Government of Canada Department of Supply and Services, File No. 17ST T80080-8-2531, for Transport of Dangerous Goods Branch, Transport Canada, Ottawa by Canuck Engineering Ltd., Calgary, Alberta.

Table II.1

Summary of the Properties of Air and the Liquid Energy Gases, Hydrogen, Methane, Propane and Butane

	Hydrogen	Methane	Propane	Butane	Air
Molecular Weight	2.00	16.00	44.00	58.00	28.90
Density (kg/m^3) at N.T.P.	0.08	0.67	1.86	2.45	1.22
Relative Density (air = 1)	0.07	0.55	1.52	2.00	1.00
Calorific Value at N.T.P. (MJ/m^3)					
a) Gross C.V.	12.10	37.70	97.90	121.80	-
b) Net C.V.	11.90	33.90	86.40	112.40	-
Stoichiometric Combustion Requirement (Volume per Volume Fuel)					
a) Oxygen	0.50	2.00	5.00	6.50	-
b) Air	2.38	9.52	23.81	30.95	-
Combustion Products: (Stoichiometric) Volume per Volume of Fuel Gas					
a) CO_2	-	1.00	3.00	4.00	-
b) H_2O	1.00	2.00	4.00	5.00	-
c) N_2	1.88	7.52	18.81	24.45	-
d) Total (Dry)	1.88	8.52	21.81	28.45	-
e) Total (Wet)	2.88	10.52	25.81	33.45	-
Burning Velocity of Stoichiometric Mixtures Initially at N.T.P. (ms^{-1})	3.20	0.33	0.33	0.33	-
Weaver Flame Speed Factor	100.00	14.00	16.00	16.00	-
Specific Heat at Constant Pressure at N.T.P. ($kJ\ K^{-1}\ kg^{-1}$)	14.30	2.22	1.66	1.67	1.00
Specific Heat at Constant Pressure per Volume at N.T.P. ($kJ\ K^{-1}\ m^{-3}$)	1.18	1.46	3.00	3.98	1.18
Critical Properties:					
a) Temperature (K)	33.00	190.00	370.00	-	-
b) Pressure (MPa)	1.29	4.62	4.25	-	-
c) Density ($x10^{-3}$) $kg\ m^{-3}$	0.031	0.163	0.220	-	-

Table II.1 (cont'd)

	Hydrogen	Methane	Propane	Butane	Air
Properties at 1 bar Pressure:					
a) Melting Point (k)	14.00	91.00	85.50	-	-
b) Enthalpy of Fusion (kJ kg^{-1})	58.00	58.50	57.20	-	-
c) Boiling Point (k)	20.30	111.50	231.00	-	-
d) Enthalpy of Vaporization (kJ kg^{-1})	448.00	510.00	231.00	-	-
e) Liquid Density at B.P. x 10^{-3} ($kg\ m^{-3}$)	0.071	0.42	0.58	-	-
Viscosity (Dynamic) (mPa.S)	8.10	9.70	-	-	16.20
Viscosity (Kinematic) ($m^2\ Ms^{-1}$) at N.T.P.	82.00	11.20	-	-	10.80
Thermal Conductivity at N.T.P. (mW $m^{-1}\ k^{-1}$)	182.00	34.00	-	-	26.10
Flammability Limits (Percent by Volume) at N.T.P.					
a) Lower Limit in Air or Oxygen	4.00	5.00	2.10	1.80	-
b) Upper Limit in Air	75.00	15.00	9.50	8.40	-
c) Upper Limit in Oxygen	95.00	61.00	55.00	49.00	-
Maximum Safe Percentage of Combustibles with Air at N.T.P.					
a) CO_2 as Diluent	5.90	14.60	14.30	14.50	-
b) N_2 as Diluent	5.00	12.10	11.40	12.10	-
Minimum Auto-Ignition (In Air) Temperatures at Pressure of 1 Bar (K)	673.00	813.00	723.00	678.00	-
Minimum Ignition Energies at N.T.P.	0.019	0.28	0.25	0.26	-
Fraction of Stoichiometric Fuel Ratio in Air for Minimum Ignition Energy	1.00	0.90	1.30	1.50	-
Minimum Ignition Energies at N.T.P. for Stoichiometric Mixture	0.019	0.34	0.30	0.38	-
Quenching Distance (mm)	0.64	2.10	1.90	1.90	-
Volumetric Expansion from Liquid at Normal Boiling Point to Gas at N.T.P.	850.00	625.00	274.00	233.00	-
Chemical Composition	H_2	CH_4	C_3H_8	C_4H_{10}	O_2+N_2

Hydrogen is flammable in air when the hydrogen-air mixture varies in the range 4-75 percent by volume. Corresponding flammability limits are 5 to 15 percent for methane and 2 to 10 percent for propane and butane.

In unconfined open-air conflagrations, hydrogen and methane burn as 'deflagrations', that is with the flame front moving at subsonic velocity through the mixture. Propane and butane on the other hand, are prone to burn by *detonation* rather than *deflagration*. The flame front in detonation combustion moves with supersonic velocity through the combustible mixture. The effects are similar to an explosion with significant blast effects on the surroundings. In confined environments hydrogen and methane can also burn in the detonation mode as well as the deflagration mode.

The controlled steady state combustion of energy gases has been studied intensively over many years and there is an extensive literature on the subject. Excellent handbooks for gas burner design and for the proper safety procedures in use have been prepared by gas industry associations and research establishments, including the Canadian Gas Association, Toronto, Ontario, the American Gas Association, Washington, D.C., the Institute for Gas Technology, Chicago, Illinois and the National Gas Board, London, England.

Liquid gases are invariably vaporized to the gaseous state prior to combustion and customarily before passing the combustor jet. Vaporization is accomplished by release of pressure or by heating (using a vaporization coil in the exhaust gases, etc.). There exists a wide variety of different burner types and systems and no attempt has been made here to classify and select the preferred system for rail locomotive application. Most burner systems attempt to achieve effective mixing of the gas fuel and combustion air prior to ignition so as to result in a stable combustion pattern with maximum release of energy in the smallest volume of combustion space. Fuel air mixtures are customarily near stoichiometric proportions with a relatively slight excess air supply. Gases can be readily combusted in fluidized bed systems as well as the more customary jet burners. Fluidized bed combustion systems operate at much lower temperature than the near-adiabatic flame temperatures of a well-designed stoichiometric jet burner system. However, heat transfer rates to a cold body immersed in the fluidized bed can be as much as an order of magnitude greater than convective heat transfer rates from high temperature but very low density products of combustion.

12.4 Availability of Liquid Energy Gases

Methane, ethane, propane and butane are all produced in large volumes by field recovery from 95 gas processing plants (principally in Alberta) and from 28 refinery operations located all across Canada. The methane is used both as a chemical feedstock and as the principal Canadian fuel for space heating of buildings. Substantial quantities are exported to the United States. Butane and ethane are principally consumed as chemical feedstock.

Propane is produced in large volume (125,000 barrels per day in 1978 and projected to continue about that level to the year 2000). About 40 percent is used domestically for vehicle propulsion, space heating and as a chemical feedstock. 60 percent is exported. Propane presently exists

in abundance and more could be produced if there was sufficient demand for it. Most propane is produced in Alberta. Of 28,000,000 barrels moved in 1978, 57% was transported by pipeline, 42% by rail and 1% by truck transport. Most of the propane shipped by rail or pipeline is eventually delivered from distributor terminals to consumers by truck. There already exists a virtually nationwide propane distribution system and substantial accumulated experience in its use. The railways are accustomed to transporting propane in pressurized tank cars and also to using it for a variety of purposes.

At present there are no regular large-scale shipments in Canada of liquid natural gas (LNG), principally methane, or liquid hydrogen (LH_2) by any mode of transport. An LNG satellite plant at Three Rivers, Quebec, is supplied from the Gaz Metropolitan facility at Montreal by one road tanker making about 80 trips a year to transport a total of 2 Mm^3 LNG per year. This will be superseded when the Trans-Quebec-Maritimes pipeline is complete and operational.

Various proposals for importing LNG to Atlantic Canada from the Arctic Islands and overseas are at different stages of development. All of these appear to anticipate vaporization of the gas at the tidewater terminal for subsequent delivery to vapour-phase gas transmission lines. Both road and rail transport of LNG appears feasible in 'super-insulated' cryogenic tank cars or trucks. Road transport of LNG is routinely accomplished for the Three Rivers, Quebec satellite facility mentioned above. Earlier, a similar facility at Squamish, British Columbia operated for some time but was abandoned because of the high costs involved. Road transport of LNG from Boston Harbour to Lowell, Massachusetts is also well established but so far as is known there has been no rail transport of LNG in North America.

The only regular user of LH_2 in Canada is Bell Northern Laboratories of Ottawa who receive 7000 liters a month in portable stainless steel dewar flasks from the United States. A liquid hydrogen plant having a capacity of 15 tons is under construction at Sarnia, Ontario and others are foreseen. Transportation of LH_2 by ship or barge is prohibited by international agreement but is routinely transported by super-insulated railway tank car and by highway tanker trucks in the United States.

12.5 Hazards of the Liquid Energy Gases

The principal hazard involved with all the liquid energy gases is the very high probability of fire and explosion following a spill of the liquids.

Major liquid gas spills can arise as the result of the structural failure of the storage tank by rupture in collision or by collapse of the tank due to structural failure or over-pressure. Minor spills can arise as the consequence of improperly executed loading, unloading or over-pressure venting procedures.

Liquid hydrogen vaporizes readily and hydrogen is so light compared to air (molecular weight is 2 compared with 29) that the gas rises and rapidly dissipates quickly into the atmosphere provided there is no confinement of the gas. However the flammability limits of hydrogen-air

mixtures are so wide and the ignition temperature and energy requirement so low that a very high probability of combustion exists for a short interval after the spill occurs.

Methane spills evaporate less readily than hydrogen. On land there is an initial flash of evaporation when the cold liquid first hits the ground surface. Soon, however, the ground in contact with the spill becomes cooled and the rate of evaporation settles down to a steady value. The cold dense vapour produced does not rise into the atmosphere until it is further heated. The vapour remains as a cloud close to the ground and gradually mixes with air to a flammable mixture. The liquid pool and the vapour cloud will follow the local topography and run to the lowest point. There is particular danger where the gas or liquid may enter tunnels, sewers and the like.

Propane is stored as a liquid under pressure at atmospheric temperature so an escape from the tank immediately vaporizes. However, the vapour is heavier than air (molecular weight of 44 compared with 29) and so does not dissipate into the atmosphere but remains as a heavy dense vapour cloud in the low topographical region. Again tunnels and sewers are a major concern particularly since propane burns with the explosive characteristics of a detonation combustion.

Liquid gas spills on water behave differently to spills on land. The water surface is generally flat and smooth so the liquid gas spreads rapidly over the surface. The rate of vaporization of the liquid therefore increases progressively over a wider area. Furthermore by natural convective action the cooled water at the surface is displaced by warm subsurface water to maintain the high rate of vaporization. Thus spills in water vaporize quickly to establish a flammable plume above the spill site.

A relatively minor but important hazard of liquid energy gases is the physiological hazard of asphyxiation and of frostbite as a consequence of immersion in a cryogenic liquid. Asphyxiation occurs as a result of displacement of the air by heavy vaporized gas.

Another important hazard arising during the escape of cryogenic liquid is the failure of structural materials resulting from the ductile/brittle transformation that occurs with some materials at low temperature. Mild steel experiences the ductile/brittle transition at temperatures not much below 0°C. In the brittle condition the material has virtually no impact strength and snaps like a carrot following a blow that at normal temperatures it would sustain without damage. Steel alloys containing appreciable amounts of nickel (9 percent for LNG service) and some aluminum alloys do not experience the transition and so are used for cryogenic equipment. However, in an accidental spill many materials not designed for cryogenic service may become immersed in the cryogenic liquid. Rubber and plastic elastomeric materials become hard and brittle at very low temperatures. Furthermore, all materials experience some contraction on cooling. Cooling to cryogenic temperatures can induce very high stresses in rigid structures not designed to be cooled so severely. The probability of structural failure of non-cryogenic rated steel is very high in the event of a major liquid gas spill.

Experience has shown that the use of proper procedures and equipment

for loading, unloading and transporting enable all the liquid energy gases to be transferred and used safely with no danger to life and property. Nevertheless it would be unrealistic to believe the use of these fuels for locomotives would eliminate the prospect of future railway accidents. Moreover, it is simply not realistic to design transportation equipment with sufficient armour to ensure that no spills would occur. Finally the great virtue of the liquid gases, their compelling affinity for oxygen with consequent energy release makes it virtually inevitable that ignition and conflagration of the gas will follow a spill. There is no question that liquid energy gases pose more hazards than the use of diesel fuel, perhaps of the same order as those associated with the use of gasoline.

Accidents are minimized by the use of trained experienced personnel. An excellent training course for drivers of liquid petroleum gas trucks has been developed by the Southern Alberta Institute of Technology, Calgary, with the support of the Propane Gas Association of Canada. This training course is available publicly at very low cost and could well serve as a model for similar training manuals for liquid hydrogen and liquid natural gas handling procedures. Similar training courses should be developed for the railway and other emergency measures personnel likely to be called upon in the event of an accident involving the liquid energy gases.

13. GAS-FIRED STIRLING ENGINES

13.1 Introduction

Stirling engines operate with any form of heat supplied to the heater at high temperature regardless of the source. Gas is a particularly favourable fuel for a combustion heated system. All the conventional energy gases, methane, propane, hydrogen, etc. burn readily with substantial release of energy in relatively low cost, uncomplicated, combustion systems

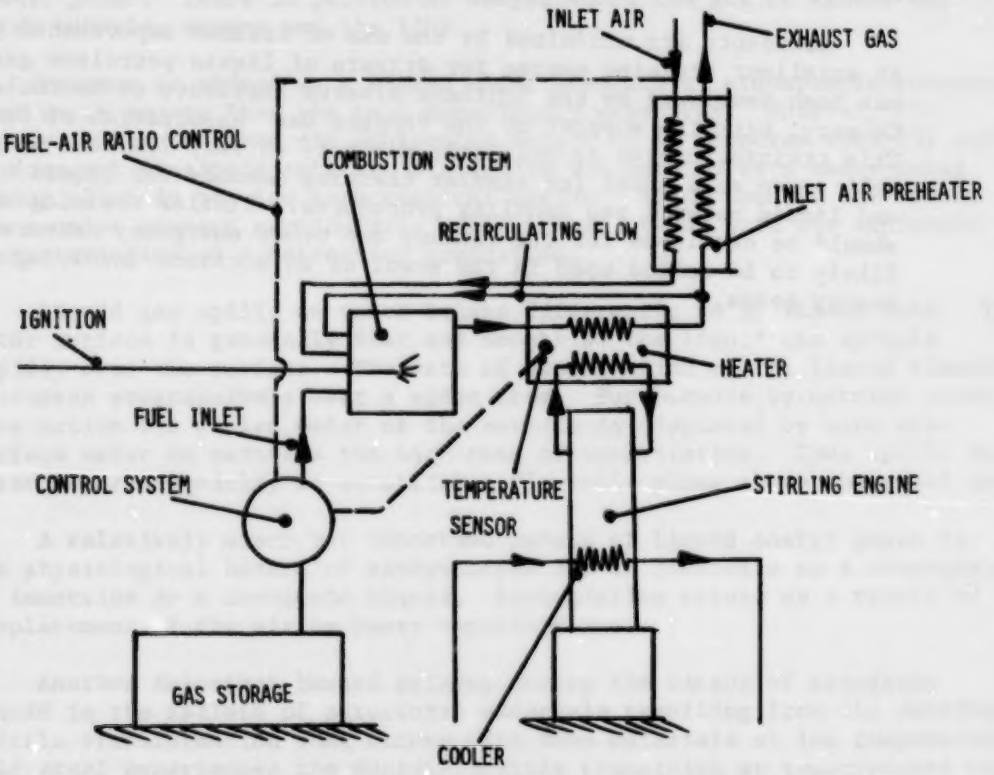

Figure 44 Elements of a Gas-Fired Stirling Engine System.

with simple controls. Under properly controlled conditions combustion is complete with virtually no exhaust residues and no hydrocarbon pollutants. Recirculation of some fraction of the exhaust products reduces the maximum combustion space temperature and minimizes the production of nitrous oxides. Air pollution is therefore negligible. Fuel gas can be stored under pressure at atmospheric temperature (propane) or at low temperature and atmospheric pressure (methane and hydrogen). The technology for fuel gas production, storage, transport, use and disposal is highly developed and the gases are relatively abundant and low in cost.

Optimum conditions for fuel gas use call for continuous combustion at near atmospheric pressure in a chamber with hot walls, conditions which are readily achieved in Stirling engine systems.

13.2 Elements of a Gas-Fired Stirling Engine System

The essential elements of a gas-fired heater system for Stirling engines is shown in Figure 44. It is comprised of a combustion chamber wherein the fuel gas and air, in approximately stoichiometric proportion, combine in a controlled combustion reaction with consequent release of heat at high temperature say 2500 K.

The hot products of combustion pass to the Stirling engine heater. In this heat exchanger thermal energy is transferred from the combustion products to the working fluid of the Stirling engine through the walls of the heat exchanger at a temperature of 1000 to 1100 K. The heater, a crucial unit of a Stirling engine, is considered in detail below.

In the heater the combustion products are cooled to say 1600 K and thence pass to exhaust. However to improve the overall system efficiency it is necessary to incorporate another heat exchanger in the exhaust duct to heat the incoming air *en route* to the combustion chamber. Depending on the design, size, cost and effectiveness of this heat exchanger the inlet air may be preheated from the incoming temperature of 280 K to say 1200 K while the exhaust gas is cooled from 1600 K to say 600 K. Provision of this heat exchanger significantly reduces the amount of fuel gas required to raise the temperature of the inlet air to that of the combustion products. It is essential in any Stirling system where overall system thermal efficiency is of importance and can be of the recuperative or regenerative type. Gas-turbine automotive type regenerative 'thermal wheel' heat exchangers are attractive for this application although most examples to date have been of the recuperative form.

Some fraction of the partly cooled products of combustion leaving the heater may be recirculated back to the combustion chamber to minimize the maximum temperature and hence NO_x production. A high temperature fan is necessary to accomplish the recirculation if it is carried out upstream of the inlet air preheater. To avoid the cost and power requirement of the high temperature fan the combustion products may be mixed with the incoming air downstream of the inlet air preheater. This, of course, has the disadvantage that the size of the inlet air preheater must be increased since it must effect the cool down and reheat of the recirculated exhaust gas fraction. Often the fraction of gas recirculated is substantial (as much as 40 to 50 percent of the total flow) so this trade-off of heat exchanger versus high temperature recirculation fan requires careful evaluation.

Combustion and heat exchange in the heater and inlet air preheater all occur at substantially atmospheric pressure. However, there are always some frictional pressure drops associated with air/exhaust gas movements and so a fan or other air moving device is necessary. When the fan is applied to the inlet air duct, upstream of the inlet air preheater, the system is said to be a forced-draft unit. The air pressure throughout is somewhat higher than atmospheric. When the fan is applied to the exhaust duct, downstream of the inlet air preheater, the system is said to be an induced draft unit and the pressure is somewhat below atmospheric throughout.

Variable load demands imposed on the engine will necessitate regulation and control of the fuel or fuel/air supply to the combustion chamber. This

is best accomplished by means of a temperature sensor attached to the Stirling engine heater. The control system is activated by the temperature sensor and seeks to maintain the heater temperature constant at some predetermined level influenced largely by the metallurgical qualities of the heater materials.

Of course the thermal inertia of the system is so high that simply reducing the fuel supply would not instantaneously reduce the engine output. Therefore additional controls having more direct influence on the engine are necessary. There are a variety of ways to achieve rapid engine response to sudden load changes and are covered in detail elsewhere (Walker 1980).

Starting and ignition systems must be included in the gas-fired heating system. With gas systems pilot flame, catalytic reaction and electric spark ignition systems are all possible and well developed commercially available units would likely be incorporated. Starting systems involve time sequenced operation of opening gas stop valves, driving fans, automatic ignition and then, after warm-up, engine starting. Prototype automotive Stirling engines made by the Philips and United Stirling companies achieve a 15 to 17 second start capability from the instant of initiation to driveaway. Larger engines would likely take appreciably longer to start.

13.3 Some Aspects of Stirling Engine Heater Design

The heater is a heat exchanger provided to facilitate the transfer of heat at high temperature from the products of combustion to the internal working fluid of the engine.

Heaters can be a) directly heated or b) indirectly heated.

Directly heated systemsare located in the flow path of the hot products of combustion so the exhaust gas pass directly over or across the tube walls or fins of the structure containing the working fluid at high pressure. The process of heat transfer then embraces:

a) convective heat transfer from the hot exhaust gas to the heater walls,
b) conductive heat transfer through the heater walls from the hot outer surface to the inner surface,
c) convective heat transfer from the inner surface to the engine working fluid,
d) appreciable radiation heat transfer may or may not be involved to the outer walls of the heater depending on the design of the combustion space.

In grossly simplified fashion the above heat transfer processes may be described mathematically by the following equations:

a) External convective heat transfer from the combustion products to the heater walls

$$(\dot{Q}/A)_e = h_e (T_g - T_e)$$

where

$(\dot{Q}/A)_e$ = convective heat flux per unit surface area for heat transfer,

h_e = convective heat transfer coefficient,

T_g = temperature of the exhaust gases,

T_e = temperature of the external surface of the heater walls.

b) Conductive heat transfer through the heater walls

$$(\dot{Q}/A)_c = -\frac{k}{x}(T_e - T_i)$$

where

$(\dot{Q}/A)_c$ = conductive heat flux through the heater walls per unit surface area of the wall,

k = thermal conductivity of the heater wall,

x = thickness of heater wall,

T_e = temperature of the external surface of the heater,

T_i = temperature of the internal surface of the heater.

c) Convective heat transfer from the heater walls to the working fluid of the engine

$$(\dot{Q}/A)_i = h_i (T_i - T_f)$$

where

$(\dot{Q}/A)_i$ = convective heat flux per unit surface area for heat transfer,

h_i = convective heat transfer coefficient,

T_i = temperature of the internal surface of the heater walls,

T_f = temperature of the working fluid in the heater.

To a first approximation:

$$(\dot{Q}/A)_e = (\dot{Q}/A)_c = (\dot{Q}/A)_i$$

or

$$h_e (T_g - T_e) = \frac{k}{x}(T_e - T_i) = h_i (T_i - T_f)$$

The maximum heater wall temperature, T_e, must be limited to the so-called 'metallurgical limit' of the material from which the heater is constructed. This is typically 1000 K for stainless steels and somewhat higher for the special alloys. Substantially higher temperatures are foreseen with the use of ceramics but silicon nitride heater assemblies for Stirling engines remain at the development stage.

The internal wall temperature of the heater will be somewhat less

(say 50-100 K) than the external wall temperature depending on the thermal conductivity of the material used and the thickness, x, of the wall. Clearly it is advantageous to minimize the thermal resistance by selecting the ratio (k/x) to be as high as possible. The thermal conductivity of metals is relatively high varying from a maximum of 400 W/m°C for pure silver and copper to a typical value of 16 W/m°C for stainless steel. The high temperature strength properties of the heater wall material and the geometry of the flow system affect the wall thickness and obviously should be made as small as possible to maximize the (k/x) ratio. The magnitude of the temperature differences $(T_g - T_e)$ and $(T_i - T_f)$ are clearly functions of the convective heat transfer coefficients h_e and h_i so that

$$\frac{(T_g - T_e)}{(T_i - T_f)} = \frac{h_i}{h_e}$$

The temperature of the hot gases leaving the combustion chamber is quite high, say 2500 K so that

$$(T_g - T_e) \approx (2500 - 1000) = 1500 \text{ K}$$

The thermal efficiency of the engine is directly related to the maximum fluid temperature T_f and so this should be made as high as possible. Assuming T_1 to be 950 K then perhaps $(T_1 - T_f)$ could be in the range (950-900) = 50 K. Then the ratio

$$\frac{h_i}{h_e} = \frac{T_g - T_e}{T_1 - T_f} = \frac{1500}{50} = 30$$

Such a range is quite possible for convective heat transfer coefficients, h, vary widely from 1 $W/m^2°C$ in natural convection to 10^5 $W/m^2°C$ for condensing water systems.

For forced convective heat transfer without phase change the heat transfer coefficient may be estimated from an equation of the form:

$$Nu = C\, Re^m\, Pr^n$$

where

- Nu = Nusselt number, (hx/k),
- Re = Reynolds number, $(\rho ux/\mu)$,
- Pr = Prandtl number, $(C_p\mu/k)$,
- C = constant, characteristic of the flow system,
- h = heat transfer coefficient,
- x = some characteristic dimension,
- k = thermal conductivity of the fluid,
- ρ = density of the fluid,
- u = velocity of the fluid relative to the wall,
- μ = fluid viscosity,
- C_p = specific heat.

For most gases $(Pr)^n \approx 1$ so that

$$h = \frac{Ck}{x}\left(\frac{\rho u x}{\mu}\right)^{m}$$

This equation clearly shows the heat transfer coefficient is a strong function of the fluid density and the velocity with a given fluid in a corresponding flow situation. Therefore the compressed, high density working fluid moving with high velocity through heater tubes or slots can be expected to have a high convective heat transfer coefficient compared with the low density, heated, atmospheric pressure, combustion products moving at moderate velocities over the outer walls of the heater tubes, although of course the actual values of the constants C, m and the characteristic dimension are not the same for internal tube flows and external

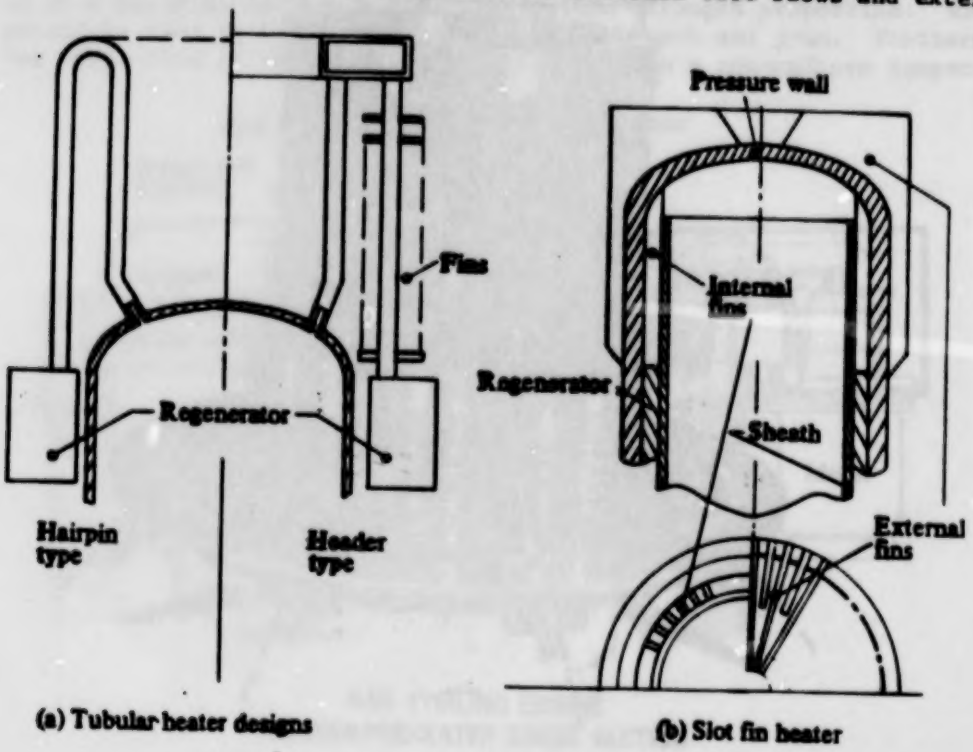

Figure 45 Types of Heater Head Design.

cross-flows. The fluid properties, conductivity, k, and viscosity, μ, do not depend much on the pressure over the range customarily used in Stirling engines.

The low value of the convective heat transfer coefficient h_g for the heat transfer between the combustion products and the heater walls is therefore accommodated by the high temperature difference $(T_g - T_e)$ between the products of combustion and the heater wall. Heater tubes with external fins may also be used to increase the area and so enhance the heat transfer from the combustion products to the heater.

These considerations lead frequently to heater designs similar to those shown diagrammatically in Figure 45. Heaters are usually of two types:

a) tubular heaters with multiple fine bore tubes closely spaced, coupling the expansion space to the regenerator,
b) finned heaters with fine internal and external slots mitted into the cylinder heat.

Two types of tubular heater are illustrated in Figure 45a. The 'hairpin' type at the left consists of nests of identical tubes bent in a U shape.

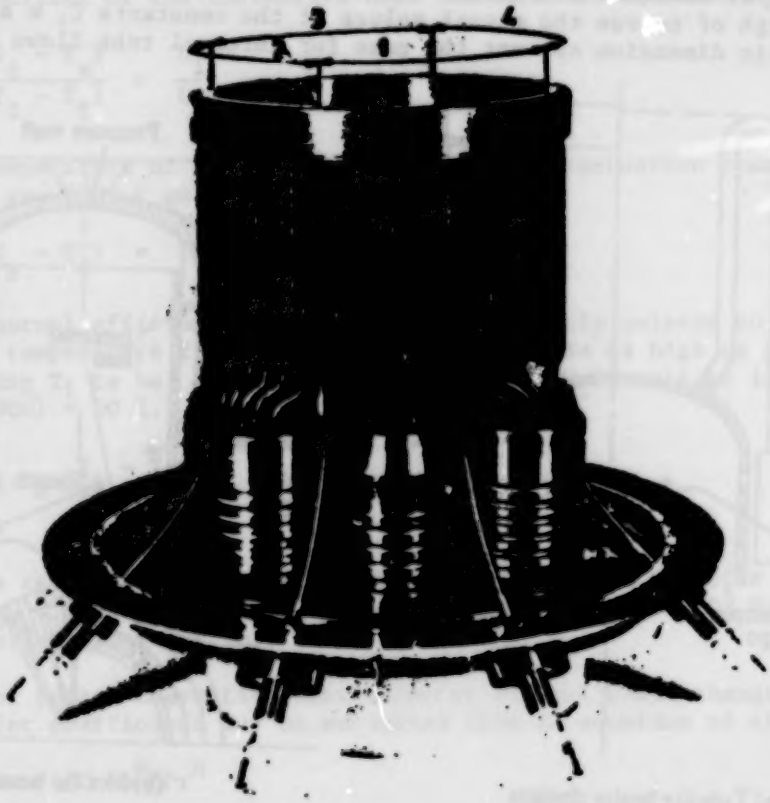

Figure 46 Typical Tubular Heated Head Assembly.

The header type consists of two concentric rings of tubes welded or brazed to an upper distribution header. This design permits the use of different diameters and types of tube for the internal and the external rows. In the example shown the internal tubes are not formed whereas the outer tube row is finned. An example of this header design is shown in Figure 46.

In tubular designs such as shown in Figure 45a it is customary to locate the combustor along the axis of the engine cylinder so the combustion space is the space enclosed by the ring of tubes. The inlet air

preheater may be of annular form surrounding the heater tubes as shown in Figure 47. The products of combustion pass from the combustion space in cross-flow across the tubes and through the gaps between the tubes and thence enter the fine passages of the air preheater before leaving the system via the exhaust stack.

Tubular arrangements of the type shown in Figure 46 are difficult and costly to fabricate, usually by means of a dip brazing or vacuum furnace brazing process. The tubes are exposed continuously to the maximum temperature and sustain a high internal pressure. Therefore they must be of a material with good high temperature strength properties. Such materials tend to be costly and difficult to work and join. Furthermore the cross-flow of hot exhaust products leads to a non-uniform temperature

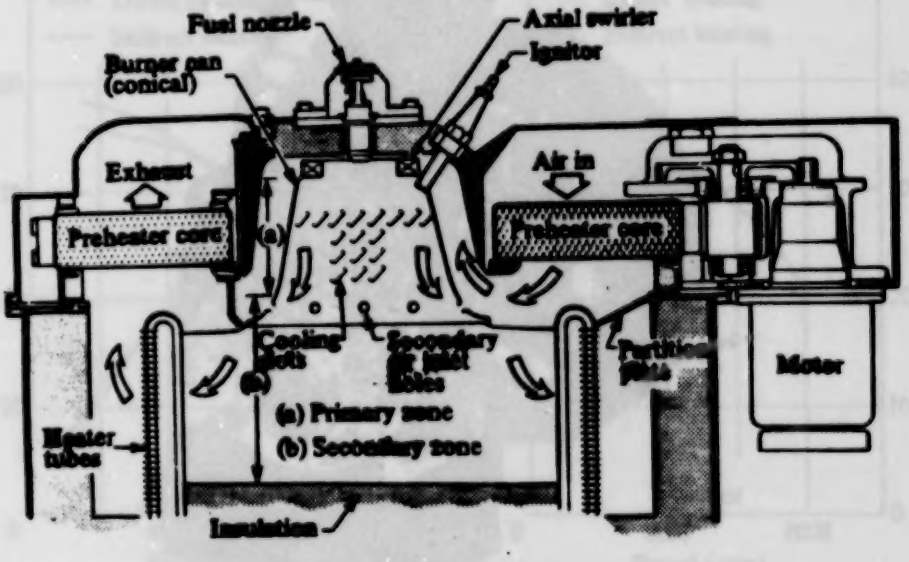

Figure 47 Hot-Parts Assembly of a Stirling Engine.

distribution around the tube. The upstream surface facing the combustion space becomes hotter than the downstream back face. Consequently there are differential thermal expansions between the upstream and downstream faces. Heater tube assemblies commonly fail because of thermal cracking induced by the differential thermal stresses and strains imposed on an already arduous stress environment. Another problem arising from thermal strains is that the interstices between the tubes become uneven with consequent maldistribution of flow across the tubes, further exacerbating the uneven temperature distribution.

A great deal of development work on tubular heaters has been done by United Stirling of Malmo, Sweden, to overcome the problems of differential thermal expansions. Some details of this development were given by Walker (1980) and references for further study are included there.

Heater designs of the finned type shown in Figure 45b are customarily found on smaller engines. This design lends itself well to an annular arrangement of heater, regenerator and cooler around the engine cylinder. An example of this type of engine is shown in Figure 48. The use of finned heaters is not, of course, necessarily confined to the annular cylinder head design and many variations and alternatives are possible. In general, internal fins or slots of the fine geometry necessary for high density, high

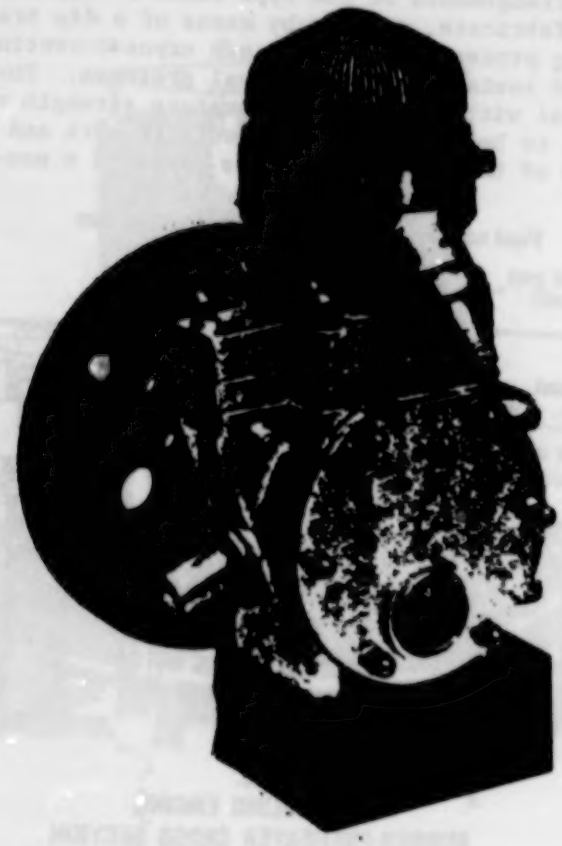

Figure 48 Single Cylinder Engine with Finned Cylinder Head Heater.

velocity fluids tend to be expensive and not simple to produce.

Burners for finned cylinder head engines are customarily arranged as an annular burner around the lower of the cylinder head with the products ascending over the fins and escaping to an axial duct leading to the inlet air preheater.

13.4 Indirect Heating

In all the above we have considered gas-fired Stirling engines with direct heating of the heater walls by the hot products of combustion. The alternative we now consider is an indirectly heated system wherein an intermediate heat transfer fluid loop is interposed between the hot

products of combustion and the engine heater. The intermediate heat transfer loop could be a gas-filled (helium) or a liquid metal (sodium, potassium sodium) heat pipe. The liquid metal heat pipe mode of operation has great advantage in simplicity, economy and size so the circulating gas system will not be considered further.

Indirect heating has several advantages over direct heating.

1) The heat source (in this case the combustion space) can be physically remote from the engine. This was the principal advantage of the heat

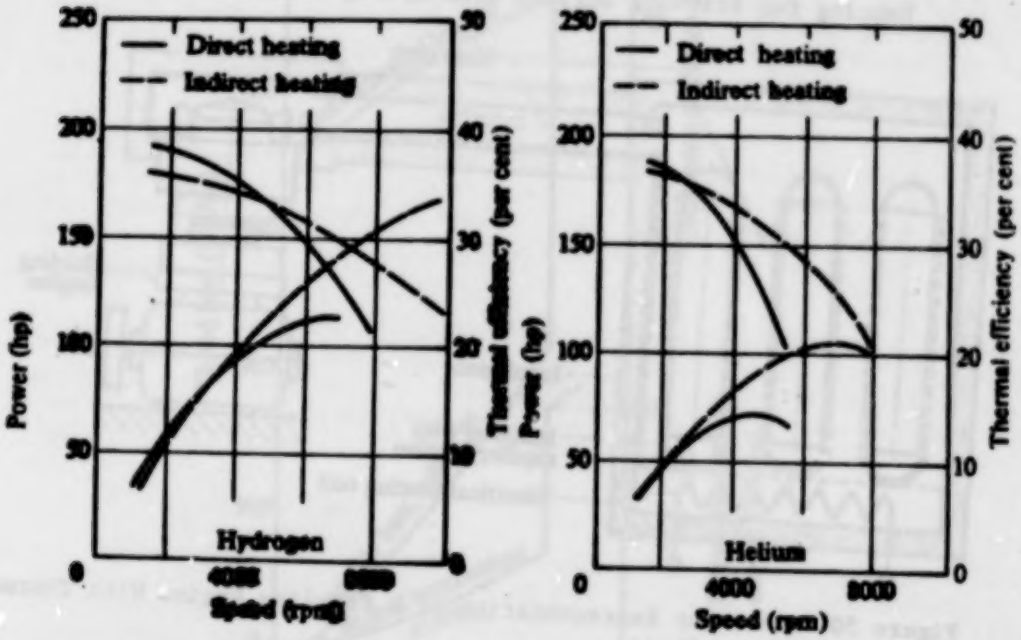

Figure 49 Comparison of Engine Performance with Direct and Indirect Heating.

pipe considered in earlier work (Part I) where the fluidized bed coal combustion was located beneath the locomotive frame between the trucks.

2) The heat pipe can act as a thermal flux transformer wherein the heat is received (from the low pressure low density exhaust gases) at a low thermal flux and over a large area but is delivered (to the engine heater tubes) at a very high flux over a small area. The rates of heat transfer of boiling and condensing liquid sodium are so high as to offer virtually no thermal resistance. This allows the design of the heater tubes/fins to be dominated entirely by considerations of internal fluid flow and conductive heat transfer in the tube/fin walls. This provides for short, small tubes, which reduces the engine dead space and thereby significantly increases the engine power output and thermal efficiency.

3) Another major advantage of indirect heating is that the condensing sodium vapour ensures a uniform temperature distribution over the whole heater external surface. The 'hot spots' which invariably occur with direct combustion heated systems are thereby eliminated. This eases the differential thermal strain problem discussed above. Furthermore, the *average* temperature of the heater system surface corresponds to the *maximum* temperature for the temperature is the same all over. With direct heating the average temperature is always as much as 100-150°C below the maximum 'hot spot' temperature which of course must be no greater than the metallurgical limit. The effect of indirect heating is therefore to elevate the working fluid average temperature with consequent improvement in both power output and efficiency.

A comparison of engine output and efficiency with direct and indirect heating for Stirling engines with hydrogen and helium was given by

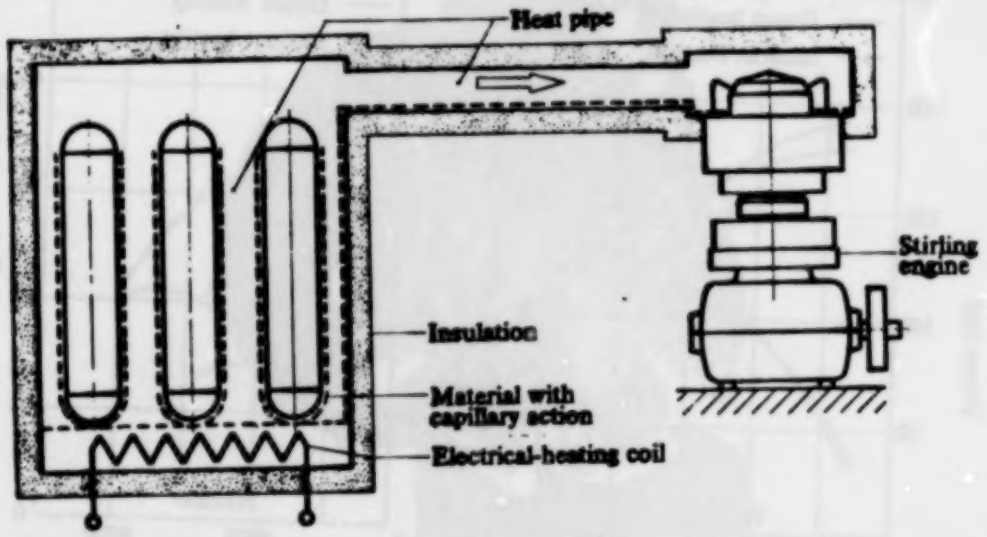

Figure 50 Schematic Representation of a Stirling Engine With Thermal Energy Store.

Meijer (1970) and is reproduced in Figure 49.

4) A further advantage of indirect heating for multiple cylinder engines is that several cylinders could be energized from a single combustion system. Thus a sixteen cylinder diesel conversion with eight Ringbom-Stirling engines might be equipped with two combustors each supplying four Ringbom-Stirling cylinders through indirect heat pipes. For low power operation only one combustor would be activated.

For the large engines necessary for locomotive propulsion the advantages of indirect heating appear sufficiently compelling to outweigh the obvious disadvantage of increased complexity and cost associated with the application of liquid metal heat pipes.

13.5 Thermal Storage Systems

Stirling engines can be energized from any source of heat. One attractive possibility that has received considerable research development attention for automotive propulsion is the combination of a thermal energy store or thermal battery, recharged overnight, and a Stirling propulsion engine energized by the thermal energy store during the day. There is no reason why this same combination of thermal energy store and Stirling engine could not be used for locomotive propulsion although no work in this direction

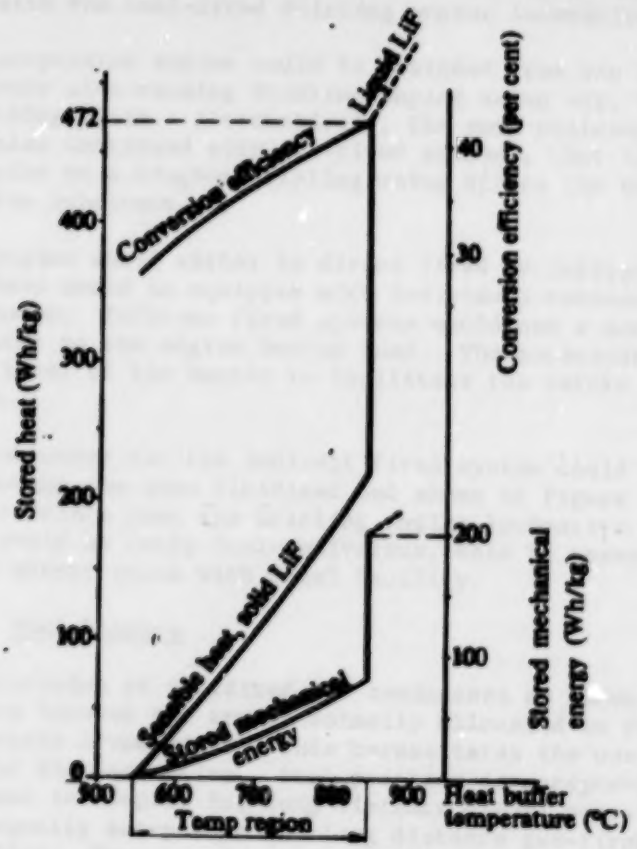

Figure 51 Energy Content of Lithium Fluoride.

is known.

The principle of the thermal energy store/Stirling engine combination is illustrated in Figure 50 (reproduced from Meijer (1970)). A favoured material for the thermal battery is lithium fluoride for it has a high latent heat of fusion at an appropriate temperature (850°C). The energy content per kilogram of lithium fluoride is shown in Figure 51. Assuming the engine conversion efficiency, also indicated in the same figure, the effective stored mechanical work can be determined. As shown in Figure 51 as much as 200 Wh/kg of effective engine work output can be stored. This is about one twenty-fifth the energy density of diesel fuel but whereas the oil has gone forever once it has been used to energize the diesel engine the

lithium fluoride remains to be reused again and again. It must be remelted and reheated but this can be done at the locomotive shed using electric resistance heating or natural gas combustion.

This novel scheme opens the way to electrify the railways without electrification or to use natural gas without the need to cool the gas to cryogenic temperatures in order to liquefy it and to provide cryogenically insulated storage facilities both onboard the locomotive and at fuel depots.

14. GAS-FIRED STIRLING ENGINE LOCOMOTIVE

14.1 Introduction

The gas-fired Stirling engine locomotive would likely have many similarities with the coal-fired Stirling engine locomotive shown in Figure 42.

The propulsion engine could be designed from the start as a large comparatively slow-running Stirling engine using air, hydrogen or helium as the working fluid. Alternatively, the same philosophy espoused earlier could be also continued with gas-fired systems, that is, converting the diesel engine to a Ringbom-Stirling using air as the working fluid and water as the lubricant.

The engine would either be direct fired or indirect fired. Direct fired engines would be equipped with individual combustor systems for every cylinder. Indirect fired systems would use a sodium heat pipe coupling the combustor to the engine heater head. The combustor would likely be below the level of the heater to facilitate the return of the sodium condensate.

The combustor for the indirect fired system could take a variety of forms including the same fluidized bed shown in Figure 42. Indeed the possibility exists that the Stirling engine locomotive with fluidized bed combustor could be truly fuel-omnivorous, able to consume coal and any of the liquid energy gases with equal facility.

14.2 Fuel Storage

The provision of fluidized bed combustors as shown in Figure 42 takes up the space between the trucks normally allocated to fuel storage in diesel-electric locomotives. This necessitates the use of a fuel tender towed behind the locomotive. Such practice is acceptable for long-haul operation and is adopted for long distance diesel operation. It would likely be equally acceptable for long distance gas-fired Stirling locomotive operation. However, for short haul and yard switching vehicles the replacement of under-frame diesel fuel tanks by liquid energy gas fuel tanks would be most attractive. The calorific values per unit mass and volume and the density at appropriate pressure and temperature of diesel fuel and the liquid energy gases is given below:

Fuel	Calorific (MJ/kg)	Value (GJ/m^3)	Density (kg/m^3)	Pressure (atm)	Temperature (K)
Diesel Oil	43	35	815	1	288
Propane	50	25	512	6.6	288
Methane	55	23	423	1	112
Hydrogen	141	10	71	1	20

Comparative Fuel Properties

The above table shows that diesel oil has a comparatively low calorific value per unit mass but has a high density so the calorific value per m^3 is 40 percent greater than propane or methane and three and a half times as much as liquid hydrogen.

Now the space between the trucks of a locomotive is limited so the replacement of diesel oil tanks by liquid energy gas tanks would therefore result in a substantial reduction of locomotive power or mileage capacity between refuelling. The penalty would be even greater than suggested above because the cryogenic fuel tanks for the methane and hydrogen fuels must be super-insulated. The tank walls are therefore relatively thick and occupy perhaps as much as 10 percent of the total tank volume. The equivalent energy value of liquid methane and hydrogen is then reduced to only 59 and 25 percent, respectively, of the energy for the same volume of diesel fuel. This suggests that frequency refuelling of the liquid energy gases would be necessary but this is undesireable for the bulk storage cryogenic tanks are costly and because special procedures and equipment are obligatory for the handling and transfer of cryogenic liquids, especially flammable ones. All this indicates that main-line locomotives would almost certainly tow a fuel tender for they could not carry the fuel on board.

Rail tank cars for liquid hydrogen transport are manufactured by the Linde Division of Union Carbide Inc. in the United States. Experience has shown the venting loss in these large super-insulated tank cars to be incredibly low, of the order of 1 percent per day, even when exposed to full-sun, subtropical, climes. Rail tank cars for methane are not in current use but road transport of methane in super-insulated highway tanks is routine. The same technique of evacuated super-insulation in the space between inner and outer, cryogenic rated, aluminum tanks is used for both highway and rail tanks. There would appear to be no particular difficulty in adapting existing hydrogen tank cars to serve as the fuel tender for liquid hydrogen/liquid methane fired Stirling engine locomotives. However, the number of such rail tank cars in North America is very limited, probably less than 100 so that new construction of an appropriate capacity would be required for large scale conversion to liquid energy gas fuels.

Transport of propane under pressure and at normal temperatures is routinely carried out in rail tank cars all across Canada. There appears to be no problem in adapting existing tank cars for use as the fuel tender for propane fuelled Stirling engine locomotives.

Commonly in train operation, four or more locomotives acting under remote control are grouped at the head and in the centre of the train. Presuming the same practice to be followed with gas-fired Stirling engine locomotives it is expected that a single fuel tender could service the fuel requirements of all the locomotives in the group.

15. THE THERMAL ENERGY STORE LOCOMOTIVE

15.1 Introduction

The combination of a Stirling engine energized by a thermal energy store (of lithium fluoride) recharged overnight by off-peak electric power or natural gas combustion was discussed earlier (see 13.5). The concept could be adapted for locomotive propulsion incorporating many of the features embodied in the locomotive shown in Figure 43. The space between the trucks containing the fluidized bed combustor in Figure 43 would be replaced by the thermal energy store of encapsulated lithium fluoride contained in sodium vapour/liquid.

Meijer (1970) suggests an energy storage density of 472 Wh/kg can be achieved for a lithium fluoride thermal energy store cycled over the temperature range 550 to 850°C. Given an appropriate engine conversion efficiency this translates to a stored mechanical work capacity of 200 Wh/kg. The energy storage density of 472 Wh/kg is equivalent to 1.7 MJ/kg. This compares with the calorific value of diesel fuel of 43 MJ/kg, i.e. the energy storage capacity of diesel fuel is (43/1.7) = 25 times that of a lithium fluoride thermal energy store. However, the lithium fluoride can be constantly reused whereas the diesel fuel can be used once only.

Kneiling (1977) suggests conventional diesel-electric units on long runs with a fuel tank car attached consume 2 (U.S.) gallons of diesel oil per 1000 gross ton miles so that two power units hauling a 5000 ton train use 10 (U.S.) gallons per loaded ton mile. A 1000 mile haul and an empty return trip consumes 15,000 (U.S.) gallons of diesel fuel per round trip. Given the energy density ratio of 25:1 for diesel oil and lithium fluoride thermal energy store, a mass of 3 kg per U.S. gallon of diesel fuel, and equivalent Stirling engine and diesel engine performance it appears a thermal energy store of 150 kg per 1000 gross ton miles or 100 kg/1000 gross tonne km. The 1000 mile haul of the 5000 ton train and empty return trip would require a thermal energy store having a mass of 1,125,000 kg (560 tons - 10 percent of the total train weight). This is clearly impractical but journeys of more moderate capacity may be contemplated. Thus a 1000 ton train on a 100 mile journey would consume 2 x 100 x 3 = 600 kg of diesel fuel or could be driven from a thermal energy store of 15,000 kg (15 tons).

15.2 Regenerative Braking

Walker (1980) has discussed the concept of using a Stirling engine both as an automotive propulsion motor and as a heat pump to recover the kinetic energy in stopping or the potential energy when descending a grade. A copy of this paper is included here as Appendix 7. This dual function concept requires the use of a thermal energy store to receive energy at high temperature from the Stirling engine acting as a heat pump. The energy is later discharged from the thermal energy store to energize the Stirling engine now acting as a propulsion engine.

The same principle could, of course, be applied to locomotive propulsion

and would appear to be most attractive for locomotives engaged in short haul commuter traffic operation with relatively frequent stops and starts. Inter-city short haul and city LRT's seem well suited thermal energy storage/regenerative braking applications for Stirling cycle engines.

16. PRESSURE BALANCED, DOUBLE-ACTING RINGBOM STIRLING ENGINE: INVENTION DISCLOSURE

Contemporaneously with the execution of the gas-fired Stirling engine work the author conceived the novel arrangement of Ringbom-Stirling engine described on the attached invention disclosure. A copy of the invention disclosure is included here because of the relevance of the novel concept to the philosophy for Ringbom-Stirling development embodied in the previous sections of the report.

INVENTION DISCLOSURE

PRESSURE-BALANCED, DOUBLE-ACTING RINGBOM STIRLING ENGINE

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INTRODUCTION

This disclosure describes a novel arrangement of a Stirling engine, more particularly of the design variant known as the hybrid or Ringbom-Stirling engine wherein the reciprocating piston is attached via a kinematic linkage to a rotating shaft and the displacer is not mechanically coupled to any other element of the engine but oscillates in the displacer cylinder under the action of fluidic forces only.

The novel feature of the arrangement described herein is the combination of two Ringbom-Stirling engine systems operating in anti-phase to form a double-acting engine. The two displacers may operate separately or may be combined as the single oscillating element described below.

The advantages of the new double-acting form, compared with the well-known single-acting Ringbom Stirling form, are:

1. the forces acting on the piston running gear (cranks, connecting rods, crankshaft big end and main bearings, etc.) are very much reduced. This is particularly true where the Stirling engine system is highly pressurized and the engine crankcase is not pressurized, invariably the case with large engines. Low loading of the running gear reduces the mechanical

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friction losses and bearing loads thereby facilitating the use of unconventional lubricants such as water or gas.

2. Fluid sealing to contain the working fluid is facilitated because the piston rod seal is smaller in diameter than the piston seal of the single-acting engine.

STIRLING ENGINE

A Stirling engine is a device operating on a closed thermodynamic regenerative cycle in which the same working fluid is repeatedly compressed and expanded at different temperature levels so there is a net conversion of heat to work or vice-versa. The system may be used for power production, for refrigeration, for 'pumping' heat to a higher temperature or for pressure generation.

The elements of a Stirling engine consist of two spaces whose volumes can be varied in a cyclic manner at the same frequency but out of phase so that volume variations in the *expanion* space *lead* those in the *compression* space by approximately one quarter of a cycle, i.e. 90° or $\pi/2$ rads. The two spaces are directly coupled through a regenerative heat exchanger and two recuperative heat exchangers. In a system intended for producing power from heat supplied at superambient temperatures the two recuperative heat exchangers are termed:

a) the heater on the expansion space side of the regenerator,

b) the cooler on the compression space side of the regenerator.

These apparently simple elements can be arranged in many different ways as described by Walker (1980)*. One well-known form called the Ringbom-

*Walker, G. (1980). Stirling Engines. Oxford University Press, Oxford, 550 pp.

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Stirling engine arrangement is shown in Figure 1. In this embodiment the displacer and piston are reciprocating elements operating in separate cylinders.

The space above the displacer is the expansion space. The space *below the displacer* (in the displacer cylinder) and *above the piston* (in the piston cylinder) is the compression space. A concentric extension below the displacer operates in another cylinder and the space below the displacer extension is called the spring space or *bounce* space.

The piston is coupled by the slider-crank arrangement shown or other suitable kinematic drive to the rotating shaft at '0'. Rotation of the shaft causes the piston to move with approximately simple harmonic motion as shown by the displacement diagram A-B-C-D. The displacer is free to reciprocate in the displacer cylinder under the action of differential pressures acting upon it. As the piston moves with the motion shown by A-B-C-D, so the displacer will move with the motion shown by the displacement diagram E-F-G-H. As the elements move in their respective cylinders the pressure of the working fluid in the compression and expansion space and the heat exchanger volumes will vary in approximately sinusoidal form as shown by the pressure diagram J-K-L-M. The pressure in the bounce space will vary approximately as represented by the pressure diagram N-P-Q-R although the variation in pressure may be small if the bounce space is large.

Heat at high temperature is supplied *to* the working fluid via the heater and is rejected *from* the working fluid at a lower temperature via the cooler. The difference in the heat supplied by the heater and

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to the cooler is the engine work output. If the work output is greater than the various losses a net engine output of shaft power will be available for useful work.

The engine work output can be represented by the area of the work diagram (a combination of pressure and volume variations) shown in Figure 2. In many engines the ratio of the maximum and minimum cycle pressure, p_{max}/p_{min}, is approximately 2:1 with a mean pressure of about 1.5:1. The work output of a given engine may be increased by increasing the mean pressure. To a first approximation the increase in output as a function of pressure is linear, i.e. doubling the pressure doubles the output. This can be readily seen in Figure 2 where the enclosed area of the upper diagram is clearly much larger than that of the lower diagram. In this case the mean pressure for the upper diagram is 3 times the mean pressure of the lower diagram.

In the simple engine shown in Figure 1 it is presumed the space below the piston is unpressurized. Therefore as the mean pressure in the working space is increased the pressure forces acting on the running gear are correspondingly increased. The friction losses in the crankshaft bearings are related to the mean pressure by the equation

$$F_b = (fn)\mu\bar{p}_m A$$

where

F_b = net bearing friction losses,

μ = bearing coefficient of friction,

$\bar{p}_m$ = mean pressure of working fluid in the cylinder,

A = area of the piston cylinder.

The bearing loads and friction losses are therefore direct functions of the mean working fluid pressure.

J.A.C.K. 8/6/81

High output engines with very high pressure working fluids would thus have high bearing friction losses and heavy structural loading requiring massive elements with associated heavy inertia forces imposed during operation.

PRESSURE BALANCING

To mitigate the problems emanating from high pressures in the working space it is necessary to apply fluid pressure to the *underside* of the piston to compensate, partly or wholly, for the high mean pressures applied to the topside of the piston.

One obvious way to do this is to pressurize the engine crankcase to the minimum or mean cycle pressure. Unfortunately this is feasible only with small engines for otherwise the crankcase becomes a complicated pressure vessel of high cost and weight.

Another possibility is to use the crosshead-linkage shown in Figure 3 with the space below the piston pressurized to the level required. If the volume below the piston is made sufficiently large (by interconnecting similar spaces in adjoining cylinders for example) the pressure in the space will remain substantially constant. If the pressure is maintained at the mean pressure of the working space then the only gas forces experienced by the crankshaft bearings and the linkage elements is $(p-p_{mean})$ where 'p' is the instantaneous value of the pressure in the working space.

DOUBLE-ACTING ENGINES

Given the crosshead linkage used in Figure 3 for the pressure-balanced engine it is now a comparatively short step to proceed to a double-acting pressure balanced engine. In this arrangement the space

below the piston is used for as the compression space of a second Ringbom-Stirling system. A second displacer cylinder is added shown in Figure 4. An alternative, and in many instances preferred, arrangement is shown in Figure 5 wherein the two independent displacers are combined in a single reciprocating element.

The frequency and mode of oscillation of displacers in Ringbom-Stirling engines depends greatly on the characteristics of the gas spring. The characteristics can be modified by change in the volume of the spring space and the pressure of the fluid contained therein. Reduction in the spring space volume increases the spring stiffness and hence the speed at which the engine will operate. A speed control system incorporating a variable volume gas spring is shown in Figure 5. The piston 'A' moves in a cylinder coupled to the spring space. The piston is hydraulically supported in the cylinder by hydraulic fluid fed to the cylinder by the speed regulating control.

Calgary June 8th 1981

Read and understood: J.A.C. Kentfield 8 June '81
Professor, Department of Mechanical Engineering, University of Calgary, Calgary, Alberta, Canada.

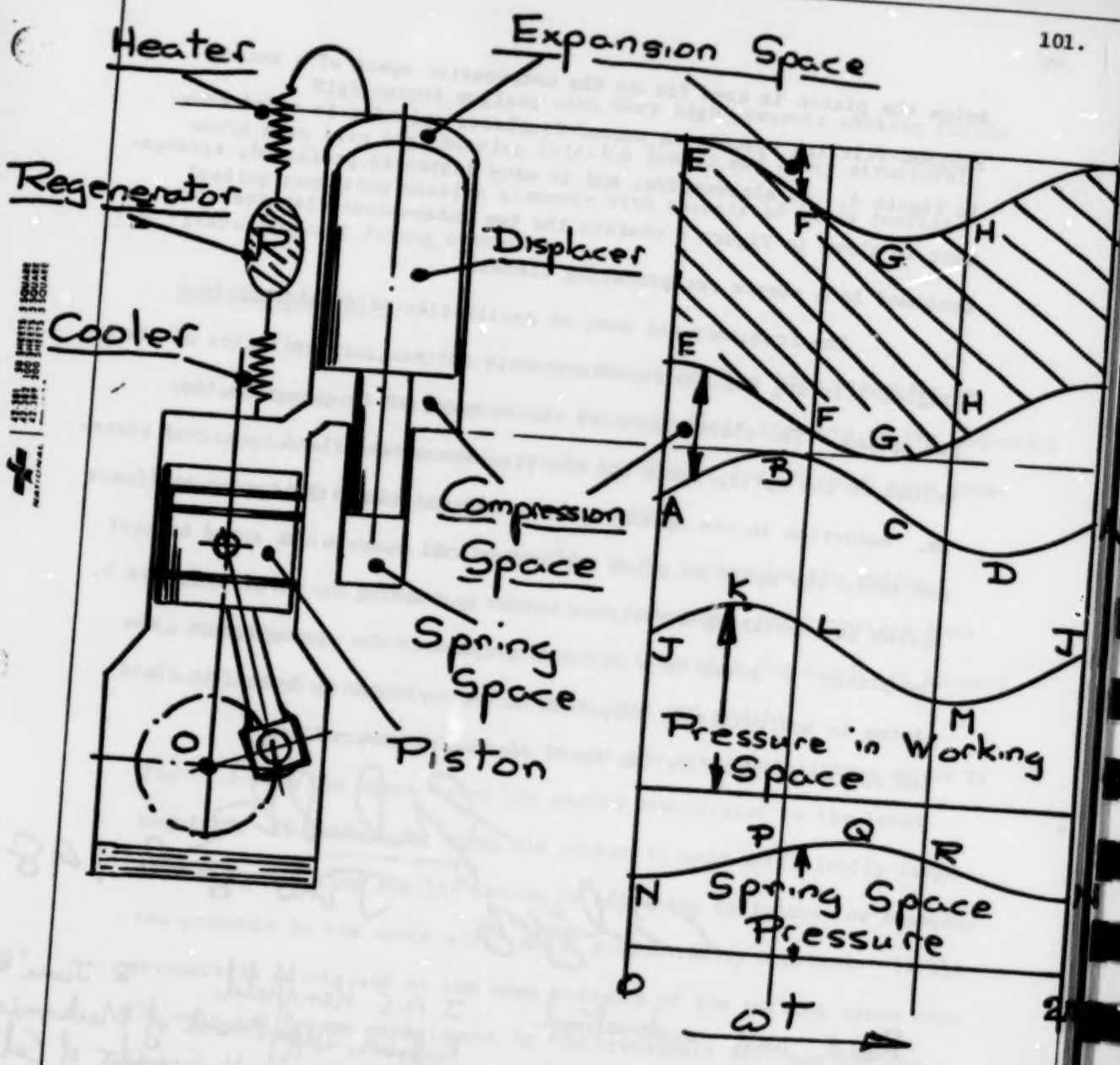

Figure 1: Ringbom-Stirling Engine

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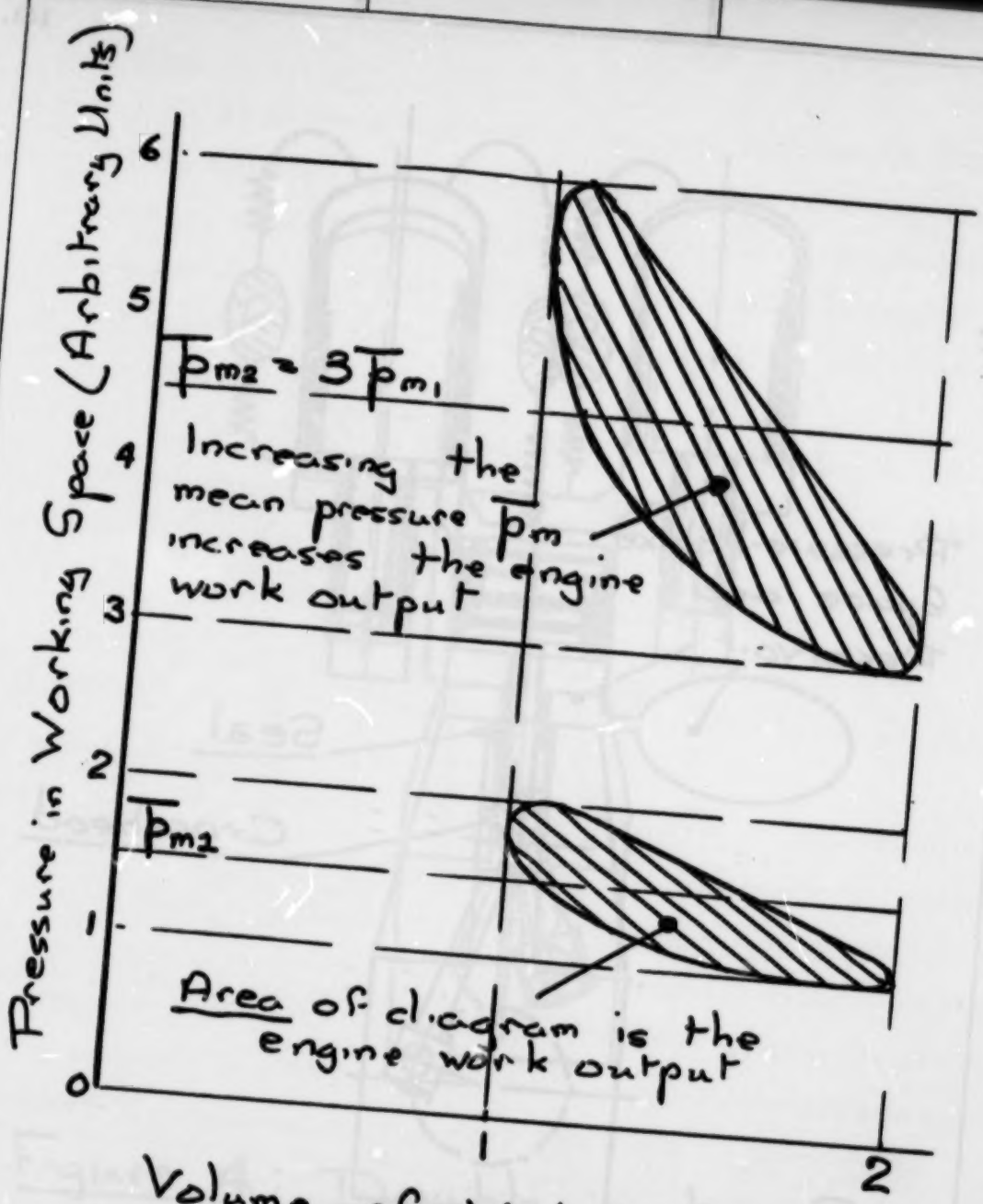

Figure 2. Work Diagram for Stirling Engine

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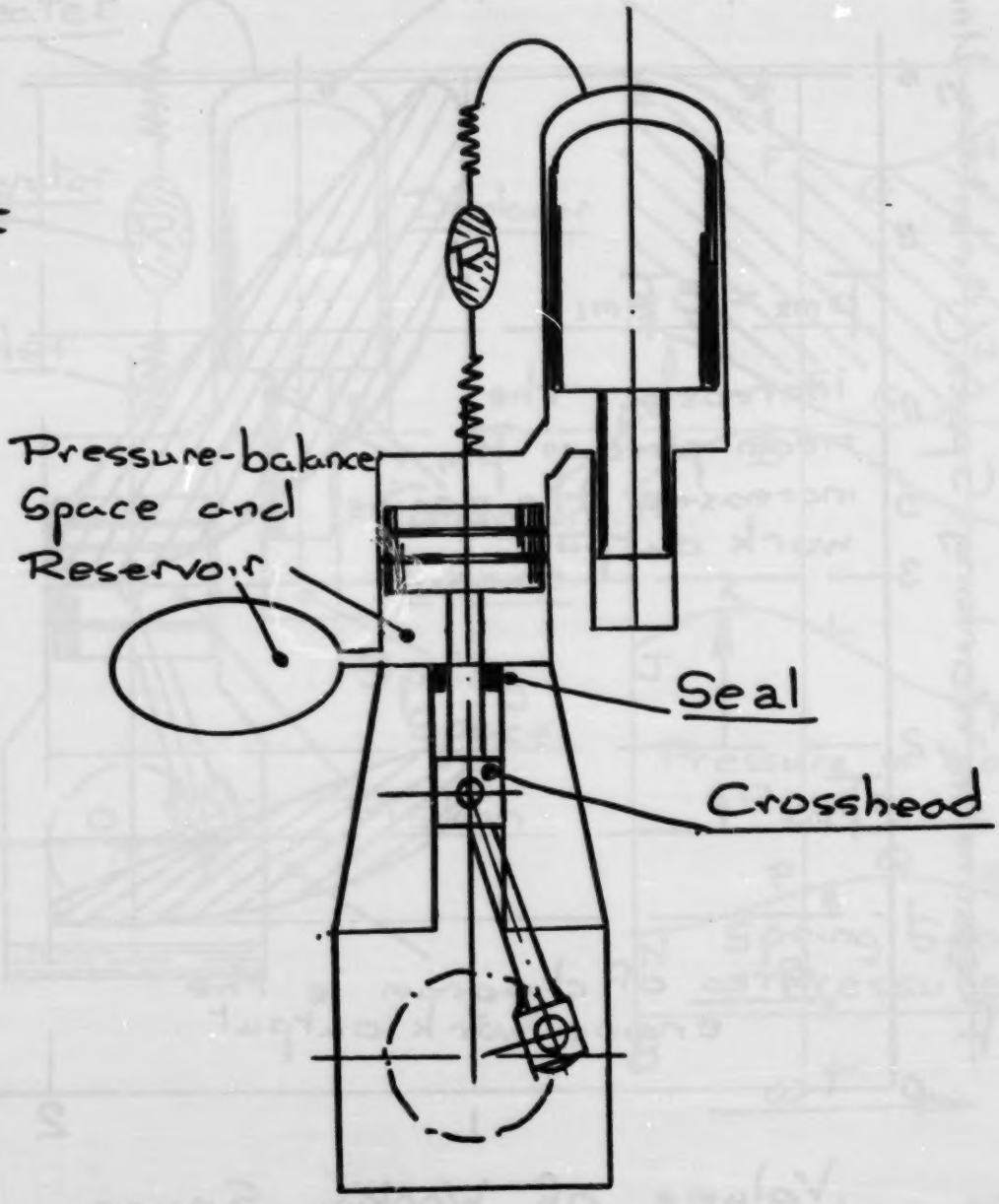

Figure 3 Pressure-balanced Ringbo

Stirling Engine

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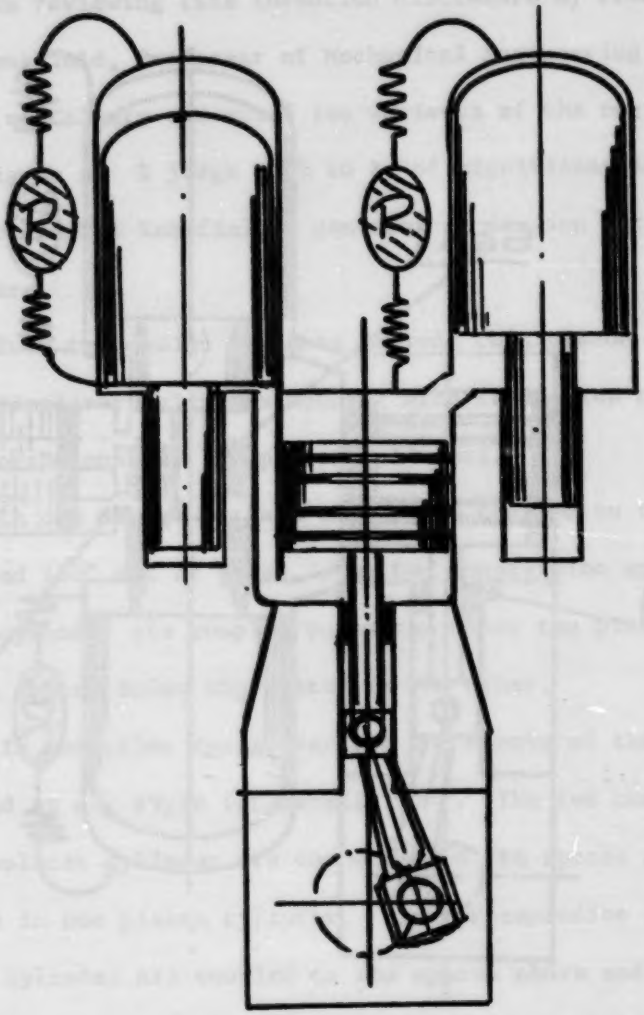

Figure 4: Double-acting Pressure Balanced Ringbom-Stirling Engine with Twin Displacers

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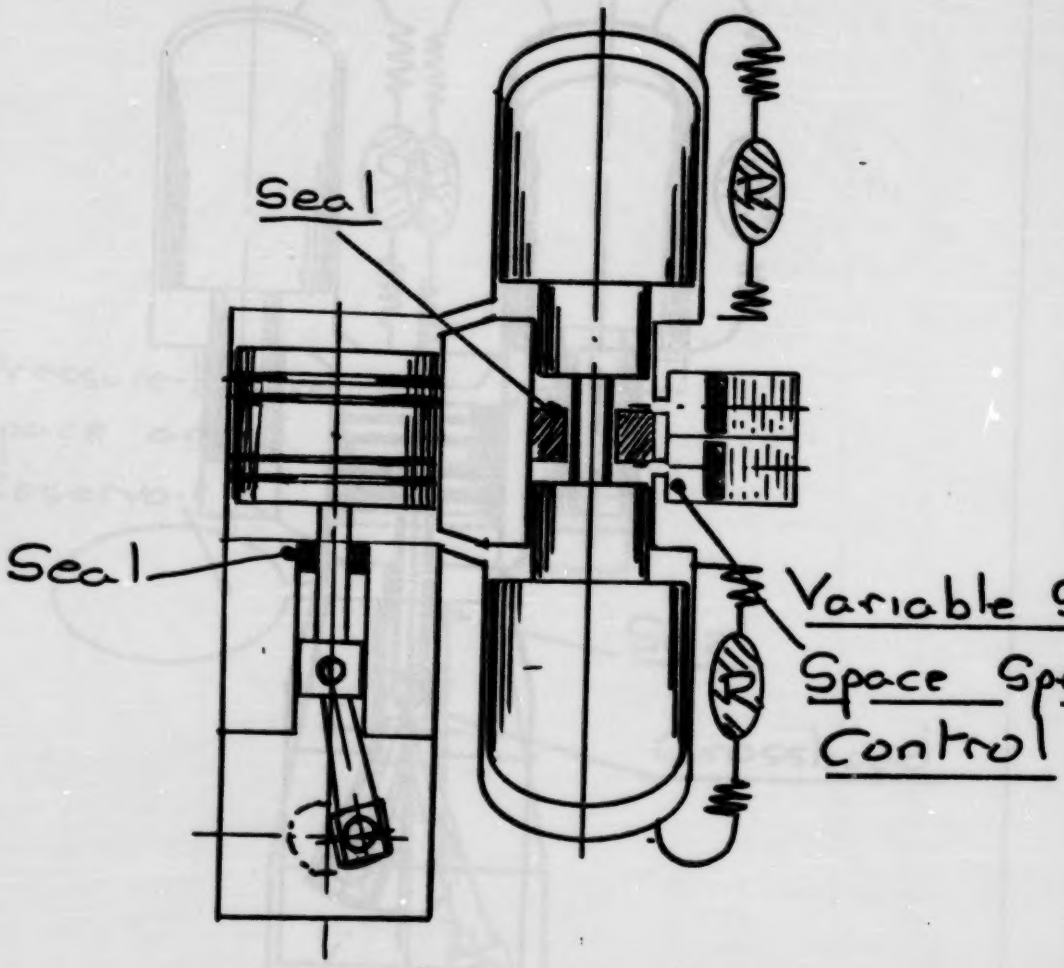

Figure 5. Double-acting, Pressure-balanced Ringbom-Stirling Engine with Dual Displacer Element.

In reviewing this invention disclosure my friend and colleague, Dr. John Kentfield, Professor of Mechanical Engineering at the University of Calgary conceived two variants of the engine arrangement shown in Figure 5. I judge both to be of significant interest and much appreciate Dr. Kentfield's generous permission for both to be included here.

Both the design variants include twin piston cylinders and twin dual-displacer cylinders mounted with their axes parallel to each other and orthogonal to the piston cylinders.

In one design variant the cranks coupled to the two pistons are arranged 180° out of phase. The two compression spaces of one displacer cylinder are coupled to spaces *above* the piston in one case and to the spaces below the piston in the other.

In the other design variant the cranks of the two pistons may be arranged at any angle (preferably 90°). The two compression spaces of one displacer cylinder are connected to the spaces above and below the piston in one piston cylinder. The corresponding spaces of the other displacer cylinder are coupled to the spaces above and below the piston of the adjoining piston cylinder. This arrangement corresponds to a simple duplication of the engine of Figure 5 with the advantage that cranks at 90° would permit the engine to be self-starting.

The preceding arrangement does not have the self-starting capability but provides the opportunity for complete symmetry of the coupling arrangements between the displacer and piston cylinder spaces.

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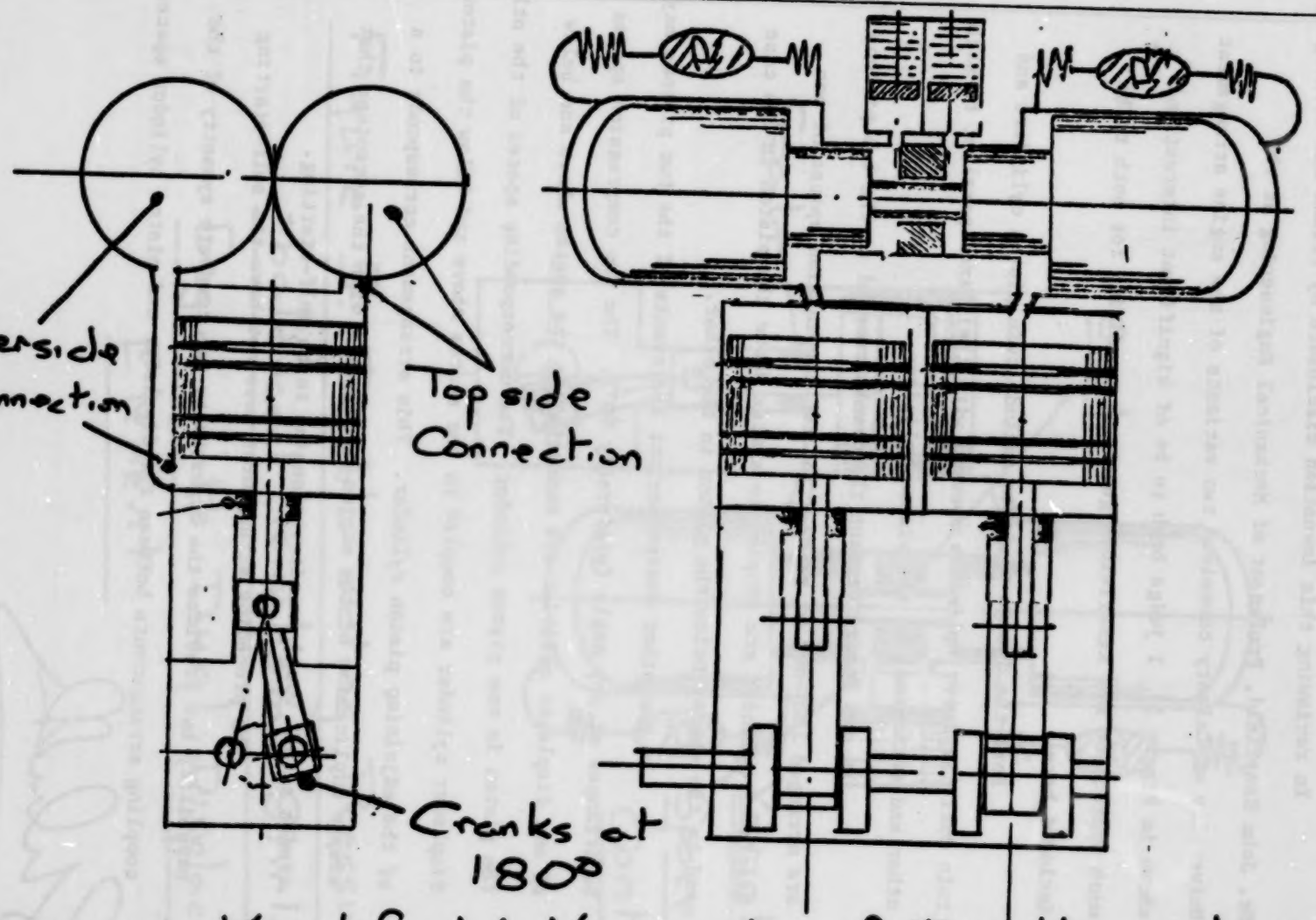

Figure 6. Kentfield Variant of Double-acting Pressur balanced Ringbom Stirling Twin-Cylinder

REFERENCES

1. (Anon, Advanced Mechanical Technology Inc.). (1980). Design and Development of Stirling Engines for Stationary Power Generation in the 500 to 3000 Horsepower Range. Report No. DOE/ET/15207-T1, U.S. Dept. of Energy, January. (Obtain from U.S. NTIS, Springfield, Va.).

2. (Anon, General Electric). (1980). Design and Development of Stirling Engines for Stationary Power Generation Applications in the 500 to 3000 H.P. Range. Report No. DOE/ET/15209-T1, U.S. Dept. of Energy, March. (Obtain from U.S. NTIS, Springfield, Va.).

3. Asselman, G.A.A. (1976). Fluidized Bed Coal Combustion as a Heat Source for Stirling Engines. Proc. Inter. Soc. Energy Conv. Conf., Lake Tahoe, Nev., (Sept.).

4. Beale, W. (1975). A Stirling Hydrostatic Drive for Small Vehicles. Proc. 10th Inter. Soc. Ener. Conv. Conf., pp. 958-960, Newark, Del.

5. Bhalla, P., Sondhi, J. and Syngal, S.P. (1973). Design Criteria for Highly Rated Diesels. Jnl. Railway Engineering, Inst. of Mechanical Engineers, Vol. 12, pp. 16-29.

6. Coward, H.F. and Jones, G.W. (1953). Limits of Flammability of Gases and Vapours. Bulletin 503, Bureau of Mines, U.S. Department of the Interior, Washington, D.C.

7. Dunn, P.D., Rice, G. and Thring, R.H. (1975). Hydraulic and Rotary Drive Stirling Engines with Fluidized Bed Combustion/Heat Pipe Systems. Proc. Inter. Soc. Energy Conv. Conf., Newark, Del., Paper No. 159141, pp. 942-947, (Aug.).

8. Ephraim, M. and Addie, A.N. (1975). Diesel Traction in North America. Jnl. Railway Engineering, Inst. of Mechanical Engineers, Vol. 14, pp. 73-84.

9. Hoagland, L.C. and Percival, W.H. (1978). A Technology Evaluation of the Stirling Engine for Stationary Power Generation in the 500 to 2000 Horsepower Range. Report No. ORO/5392-01, U.S. Department of Energy, (Sept.). (Obtain from U.S. NTIS, Springfield, Va.).

10. Holtz, R.E., Marciniak, T.J. and Uherka, K.L. (1979). DOE Stationary External Combustion Engine Programme: Status Report. Paper No. 799242, pp. 1120-1123, Proc. 14th Inter. Soc. Energy Conv. Conf., Boston, (Aug.).

11. Kneiling, J.G. (1977). Trains for Tomorrow. Trains, July, pp. 28-33, August, pp. 44-50.

12. Kukkonen, C.A. (1981). Hydrogen as an Alternative Automotive Fuel. 5P480 Alternate Fuels, Soc. Auto. Eng., Warrendale, Pa., (Feb.).

13. Meijer, R.J. (1970). Prospects of the Stirling Engine for Vehicular Propulsion. Philips Tech. Rev., Vol. 31, No. 5/6, pp. 168-185.

14. Morgan, D.T. (1978). Thermal Energy Storage for the Stirling Powered Automobile. Dept. of Energy, Report No. ANL-K-78-4135-1. (Obtain from U.S. NTIS, Springfield, Va.).

15. Michels, A.P.J. (1976). The Stirling Engine - A Study of Its Efficiency as a Function of Operating Temperatures and Working Fluids. Proc. Inter. Soc. Ener. Conv. Conf., Paper No. 769258, Lake Tahoe, Nev., (Sept.).

16. Parker, C.W. (1973). Mountain and Winter Operating Conditions in Canada. Jnl. Railway Engineering, Inst. of Mechanical Engineers, Vol. 12, pp. 6-29.

17. Rose, J.W. and Cooper, J.R. (1977). Technical Data on Fuel. 7th Edition, British National Committee, World Energy Conference, London.

18. Shigley, J.E. (1977). Mechanical Engineering Design. McGraw Hill Book Co., New York.

19. Walker, G. (1980). Stirling Engines. Oxford University Press, Oxford.

20. Walker, G. (1981). Cryocoolers. Plenum Press, New York.

21. Walker, G. and West, C. (1982). Free-Piston and Liquid-Piston Stirling Engines. (In preparation.)

22. Zabetakis, M.G. (1965). Flammability Characteristics of Combustible Gases and Vapours. Bulletin 627, Bureau of Mines, U.S. Department of the Interior, Washington, D.C.

APPENDIX 1

STIRLING ENGINE RESEARCH AND DEVELOPMENT IN JAPAN

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Japan

RESEARCH INSTITUTES

Over a period of about 20 years beginning from about 1950, over 500 units of stirling cycle air liquifiers and nitrogen generators were imported in Japan from Philips. However, it soon became clear that the time between maintenance of these machines were fairly short and that they required high level maintenance techniques. On the other hand, rapid development in oxygen steel refining by a large increase in demand for liquid oxygen in the steel industries during this period made liquid nitrogen available at a fairly low price. Due to these reasons, use of many of the machines have been discontinued.

Research on Stirling cycle machines for production of low temperatures started in Japan in 1965 by a group including the present author. Around 1970, a study program on Stirling engines as an alternative for Otto engines was initiated in connection with environment preservation problems. Table 1 lists the research establishments which are at present continuing studies on Stirling engines.

FEATURES OF STIRLING CYCLE MACHINES

Generalized T-S diagrams of Stirling cycles consisting of two constant temperature processes two constant volume processes are shown in Fig.1 according to the working temperature and the temperature span of compression and expansion.

a. Engine b. Cryogenic Prime Mover

c. Heat Pump (Air cooling, heat)

d. Cryogenic Refrigerator

A variety of other modified cycles may be conceived. In designing practical machines based on these T-S diagrams, there exists a large number of device combinations, types and shapes according to their use and capacity.

Experiments made on prototype machines of cycles a to d by the author have shown that Stirling machines possess attractive features such as high efficiency, low vibration and noise level and large torque at low revolutions. The application of the Stirling cycle to a combined engine-refrigerator system, the utilization of the cooling water to a hot water supply and heating system, a system where cold is converted to electric power by operating a cryogenic prime mover in which heat is supplied from sea water or atomospheric heat bath and rejected to LNG or liquid hydrogen heat bath has also been considered. Detailed descriptions of these applications may be found in articles published by the author (IECEC, Boston, 1979).

Although Stirling cycle machines promise potential application to a wide variety of fields, only cryogenic refrigerators are produced on a commercial basis at present. However, with progress in research towards development of practical machinces, future prospects are large for the

openning up of a wide application field.

In comparison with other heat cycle machines, the Stirling cycle machine is attractive in that it requires extreme condition technology and interdisciplinary studies. This makes its research, design and construction very interesting but also very difficult. Some of the difficulties are listed in the following:

1) Methods for maintaining highly transmissive working fluids at high purity and pressure at high or very low temperatures.

2) Methods of obtaining regenerative heating and cooling with high efficiency at high speeds.

3) The dead volume of the working fluid affects the efficiency adversely. Thus construction must be simplified to its extreme which requires studies on designing components of simplified shapes and types.

4) Proper types and organization of components still has to be established.

5) Although Schmidt's theory is very elegant, there is need to establish the fundamentals for design through proper experimentation.

6)Although any type of fuel can be used in a Stirling engine, there is need for development of high durability, low cost heater tube material and burner.

7) Besides the difficulties listed, there is still need for some innovative inventions before practical engines

may be produced.

PRESENT RESEARCH ACTIVITIES

The largest research and development effort at present is the development of a 800 ps engine for seacraft supported partially by funds from the Ministry of Transport. The development is a combined effort of universities and seacraft engine industries and is lead by Dr. H. Fujita (Meiji University).

Specifications for the engine to be developed are as follows: 720 rpm; piston diameter 220 mm; stroke 150 mm; 4-cylinder; working fluid helium; mean pressure 113 ata; heater tube temperature 700°C; double-acting, crank shaft piston driving mechanism. The main developer is the Daihatsu Diesel Co. and a one phase (the compression and expansion pistons are independent). Experiments are now being made on a 200 ps engine but the output power is still insufficient. Figure 2 shows the cross-section of the 200 ps engine.

Cycle simulation studies are being performed by Prof. N. Nakajima and Prof. H. Yoshiki's group. According to A. Asetzu, a student of Nakajima, there is good agreement between simulation and experimental results in the low revolution regime. Some results of these studies have been published in English under the title, "Pulsating Low Heat Transfer" (Bulletin of the MESJ, Vol. 8, No.3, p22-28).

Studies on a 50 kw class engine is presently being made by the author in cooperation with the Aisin Seiki Co. Ltd. It is a 4 cylinder, double acting type engine. A swash plate piston driving mechanism similar to the type developed by H. Rinia of Philips is used. The object of the present study is to clarify the characteristics and technical problems of a Stirling engine. Some results of the experiments are shown in Fig.3 to 8. These results have helped to clarify some of the engine characteristics. The cycle gas used is helium and the power shown does not include power necessary for accessories such as the blower, oil pump, etc.

Based on the results obtained, we are planning to study design methods, feasibility of practical machines and development strategies for creating original type engines.

There several other research institutes conducting studies on engines but sufficient data have not been reported up to now.

STIRLING ENGINE LOCOMOTIVE

Although the Stirling engine possesses attractive features such as possible use of many types of fuel, high efficiency, large torque at low revolution and relatively low vibration and noise, with the present state of technology, the construction cost is too high, the weight too large and there are still problems in durability.

Assuming that these problems will be overcome through future research and developments, we have invesitgated the possibility of Stirling engine locomotives in order to establish a guideline for future developments. As a first step, materials pertaining to presently available energy efficiency, estimates of efficiency and relative weight of future large scale engines, structure when used as an alternative of Diesel engines and possible influence on of technical problems to other industries were studied.

DIESEL ENGINES IN RAILWAYS At present, Diesel engines are used in locomotives, passenger cars and tractors in the Japanese railways. They are mainly used on lines connecting local towns and are gradually being replaced by electric motors. The reason for this lies in problems concerning maintenance, operation, weight, economy, noise, etc. and also matching problems with other railway lines. However, Diesel engines do have a great advantage of not requiring construction and maintenance of pickup transmission lines and electric power stations.

There are three types of machanisms for transmitting the engine output to the driving-wheel in Diesel cars presently built, namely mechanical transmission via gears, torque conversion by a liquid and electrical transmission via a generator-motor combination. The electrical method is common in most countries whereas in Japan, the liquid torque convertor is used mainly. Due to its relative

complexity, the component weights are relatively large for the electric method but has the advantage of easiness in operation. On the other hand, the liquid method is said to have the advantage of high power transmission efficiency, easiness in starting and economical operation.

In the conceptual design of a Stirling locomotive, we have considered a method to reduce the seal load for the piston ring and rod seal which is one of the problems in Stirling engines. Also in order to reduce the mechanical loss of the shaft mechanical seal, we have considered a generator type Stirling engine with a hermetric seal. For this reason, the investigation was centered on electirc type Diesel locomotives.

Table 3 shows the specification for the electric type Diesel locomotives laid by the Japanese National Railways in the 1950's. The engines used were Sulzer and MAN products.

EFFICIENCY VERSUS ENGINE OUTPUT The necessary output power of Stirling engines for use in railway vehicles and freight trains should be comparable to that of the Diesel engines currently in use, namely 1,000-2,000 HP. Estimates must be made of the efficiency and specific weight related to the output power in considering future development of Stirling engine locomotives of this scale as the operation cost is greatly affected by these factors.

In designing or operating chemical or refrigeration plants in industries, the operating condition is generally matched to the output so that maximum or optimum efficiency is obtained. The efficiency also tends to increase with increase in plant capacity. In order to clarify similar conditions pertaining to Stirling engines, a survey including other types of engines has been made. The working gas for the Stirling engines is mainly hydrogen. Available data is scarce and practical engines have not been built but our studies indicate that one may expect rated efficiencies comparable to Diesel engines in the future and further improvement up to 40% rated efficiency for large scale engines.

There are several methods for improving the efficiency. As for the heat exchanger; a) increasing the heater tube wall temperature together with devising methods for making the cycle gas temperature as near to the wall temperature as possible. Development of low price materials with high durability and sufficiently ductile is in need. According to experiments at NASA, the material 19-9 DL (HT-2) appears to be more promising than N-155 (Fig.10). b) Development of efficient coolers which cools the cycle gas down close to the envirommental temperature.

Fig.11 shows the experimental data obtained on the relation between the efficiency and the tube wall and cooling water temperatures on our small Stirling engine.

As for data on engine weight, data are very scarce. Fig.12 shows the specific weight of experimental models.

GUIDELINE FOR FUTURE DEVELOPMENTS Stirling engines of small to medium capacity have been under investigation and development for a long period and their characteristics are well defined. Technical and economical problem barring commercialization of practical engines are also well known by those who have been active in development. Future development of engines for locomotives require studies on development strategies for designing and constructing engines where the attractive features of Stirling engines are well borne out and, also, studies on methods for economical operation. Efforts must be made on removing presently known defects in course of scaling up the output capacity tenfold of that of present day models. As a guideline, the following specifications may be laid.

° For the engine, the rated power 1600 ps, hour rated power 1800 ps, revolution 1600 rpm, efficiency 40%.

° For the main generator, the rated power 1060 kw, efficiency 88 to 93%.

° Four motors, per locomotive, each with power ratings of about 250 kw and efficiency 85 to 91%.

° The speed of the locomotive over 100 km/h.

Some recommendations for future developments are as follows:

a) An electric type driving mechanism is recommended.

b) The generator is made helium tight with the same pressure as the engine crank unit (below several atmospheres). The electric power is drawn from hermetically sealed leads. This method will improve mechanical efficiency as mechanical seals are not required. Reducing the rod seal load will reduce the dynamical load of the cycle gas and improve the durability.

c) In starting the engine, the cycle gas, pressurized by a compressor which includes a DC motor, first pressurizes the crank case and generator and then is supplied to the No.1 buffer tank. When, for example, the pressure in the crank case is 3 ata, the starting pressure will be about 20 ata.

d) The minimum pressure of the cycle gas is maintained by an internal compressor or a high pressure gas cylinder.

e) If coal or firewood is used for fuel, a heat pump system for heat transport between the burner and heater using metal hydrides is considered.

f) It is desirable to operate the motor and generator at high voltage in order to make the system lightweight and to decrease the required current capacity of the hermetic terminals.

g) Utilization of the heat rejected from the engine and heat evolved in the generator for heating of the passenger car should be considered.

h) The operating temperature of the radiator is to be maintained between -40°C and 80°C by use of polyvinyl

alcohol or alcohol plus water fluid.

i) As the engine crank case and generator must be helium tight, development of lightweight and strong materials with low helium gas transmissivity is necessary.

j) Developments should be directed towards increasing the speed of revolution of the engine as in the case of developments of future Diesel engines. However, considerations should be made on the relation between the speed of revolution and voltage of the generator and motor, and the motor control system.

Fig.13 shows a block diagram of the thermal and electric system for the Stirling engine locomotive considered.

DEVELOPMENT MERITS

The development of Stirling locomotives is very significant in a country like Canada where, in comparison with other advanced nation, the average temperature is low, energy and mining resources are abundunt and where high technology (nuclear power plant, aircraft, high temperature alloys, etc.) is available. That is:

a) Characteristics of the Stirling engine such as the readiness of use of the rejected heat, the fact that the efficiency increases as the environmental temperature is lowered favourably match the Canadian climate. Good torque characteristic is favourable not only for locomotives but for medium size vessels and icebreakers.

b) The existing high technology in Canada may be used effectively in development and the experience gained in pioneering new technology should contribute to the advance in future industrial developments.

c) As the Stirling engine is an external combustion engine, abundant resources such as coal, firewood, oil, natural gas, nuclear fuel, etc. may be used as a fuel. The possible use of many alternate fuels should introduce redundancy in forming energy policies. At the same time, it will contribute in the advance of fuel utilization technologies.

In order to materialize the merits in development briefly summarized above, we believe that organizing a cooperative multination development organization consisting of members including industrial consultants from countries well experienced in special, advanced technology is preferable. This system should be more effective than the system in USA where development of Stirling engines for automobiles are being advanced by industries backed up by DOE-NASA, in fund saving, shortening of the development period and for enhancing the probability of successful achievements. This is based on the fact that the problems and difficulties barring production of practical Stirling machines for over 160 years since its invention, has recently become increasingly clear to those involved in development of a variety of machines.

We gratefully acknowledge the cooperation of Professor T. Ohtsuka, Tohoku University, Y. Kyotani, Director, Japanese National Railways (JNR) and the JNR Rolling Stock Design Division group.

DIESEL ELECTRIC LOCOMOTIVES

Locomotives presently used are steam, electric and Diesel locomotives. Research on nuclear and gas turbine locomotives have been made but are currently not in use. With advance in technology, the use of steam locomotives has decreased appreciably, but in light of recent energy problems, it has attracted some renewed interest. We believe, however, that renewed, studies should not be directed towards revision of current steam locomotives but should be studied from the standpoint of external combustion cars.

In the railways, internal combustion cars are used along with external combustion cars. Both are called power cars. At present, almost all internal combustion cars use Diesel engines and are classified into Diesel locomotives and Diesel traction cars. For transmission of the engine output power to the driving wheel, either a totally mechanical system using gears and clutches as in automobiles, a system where a fluid converter replaces the gears, a system which is also used increasingly in automobiles, a combination of the gear and fluid system, and an electric power transmission system where the engine output drives a generator which, in turn, supplies power to the trction electric motor are used. Diesel locomotives which use the electric system are commonly called Diesel electric locomotives.

The output of the engine depends on its use. Relatively small output power engines placed on connected locomotives are made so that they can be controlled from a central control board. When more than three locomotives are connected, locomotives without a driver's seat are sometimes used.

Some requrements for car Diesel engines are low fuel consumption, slow specific weight and volume per output power, easy control, high durability, high reliability, and low cost. The relative importance of each requirement depends on factors such as the type of vehicle and the line the engine is to be used on, number of units used, etc.

The electric power transmission system has the disadvantage of somewhat lower efficiency, higher cost and larger weight in comparison with the fluid system but has the advantage of relative easiness in control, especially for high power engines, easiness in introduction of remote or central control systems and also the advantage that the engine may be operated at a fixed rated output power independent of the speed of the car. Due to these advantages, the electric system is widely used in U.S.A.

The output power of the Diesel engine must match the generator output power required over the entire locomotive speed range. Insufficient engine output will result in

decrease of speed and tractive force necessary to draw the train and will cause damage to the engine. On the other hand, if the engine output is larger than that required by the generator, a governer may be used to decrease the fuel supply and present the engine from becoming overspeed but then the engine cannot be operated at its rated output power and will not be able to draw a train requiring full engine capacity. A variety of electric power transmission systems are in use. There are systems which simply allow some degree of overload of the engine. In more complicated systems, the excitation current of the generator is controlled to prevent overload of the engine or the power absorption curve of the generator is made near to a right angle hyperbola so as to match the output power of the engine with allowance for a small engine overload. There are also systems where both the excitation current of the generator and the fuel supply to the engine are automatically controlled so as to match the engine rated output power and the generator power absorption curve. In actual design, considerations must be made for power required to operate ancillary equipments and on the maximum voltage and current allowed for the generator.

With recent advance in electric cars, Diesel engine locomotives using AC motors with a variety of generator, power control systems instead of DC motor system used in

the past have appeared. Some considerations have also been made for the use of superconducting generators.

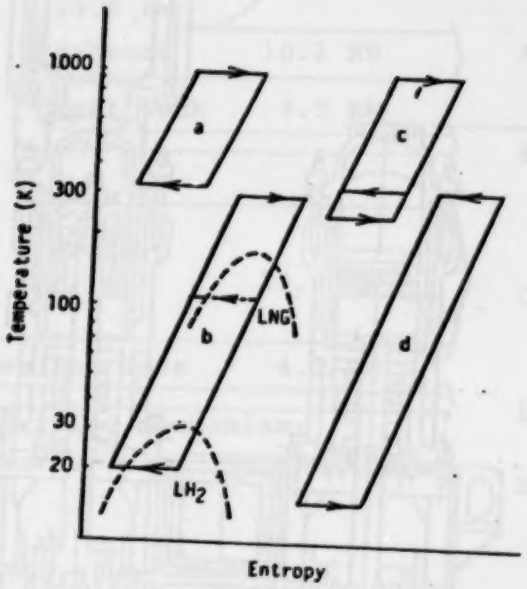


Fig. 1 T-S diagrams of the idealized Stirling cycle machines and operating temperatures

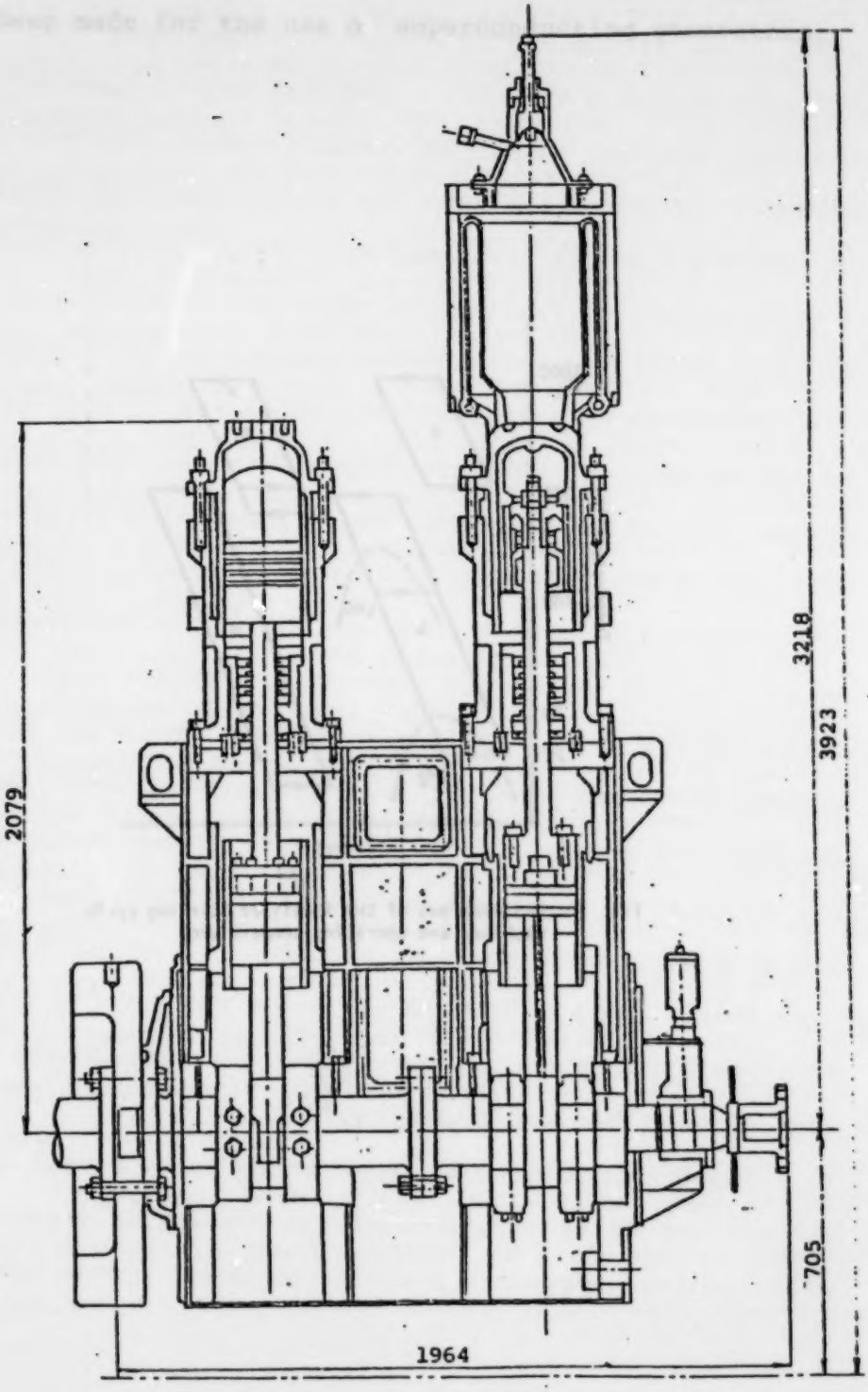
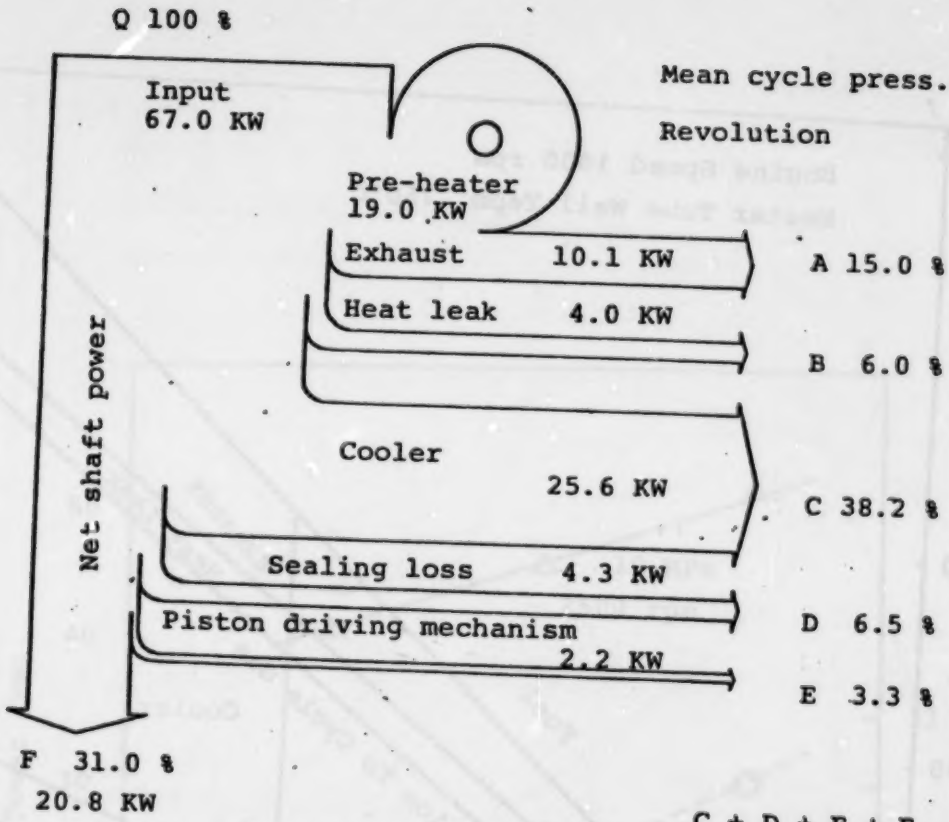


Fig. 2 Cross Section of the 200 Ps Stirling Engine



$$\text{Burner efficiency ; } \eta_B = \frac{C + D + E + F}{Q} = 79.0\ \%$$

$$\text{Indicated efficiency ; } \eta_{ind} = \frac{D + E + F}{C + D + E + F} = 51.6\ \%$$

$$\text{Mechanical efficiency ; } \eta_M = \frac{F}{D + E + F} = 76.0\ \%$$

Fig. 3 Energy Balance for the SR - 50

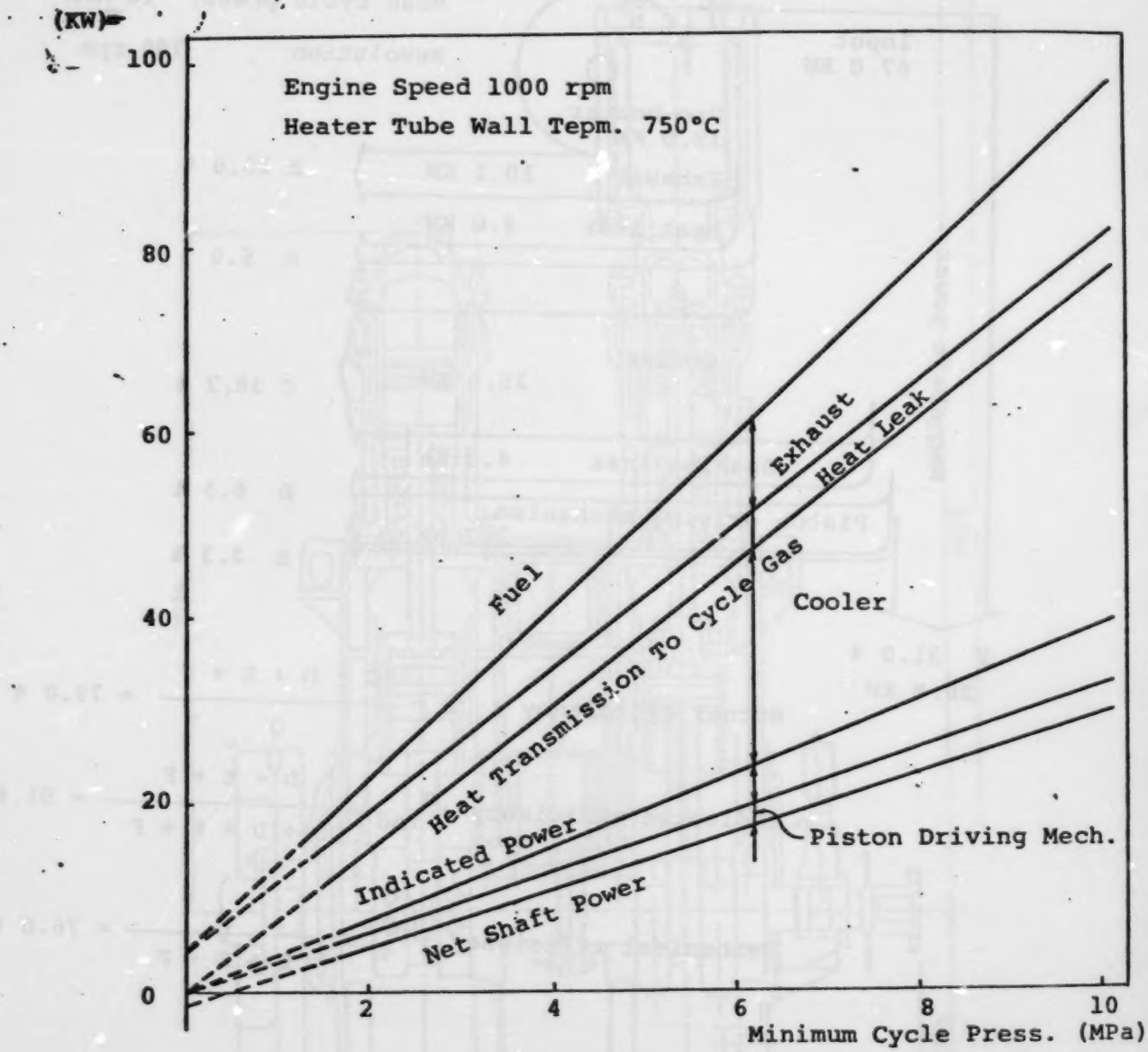

(KW)
100
80
60
40
20
0
Engine Speed 1000 rpm
Heater Tube Wall Tepm. 750°C
Fuel
Exhaust
Heat Leak
Heat Transmission To Cycle Gas
Cooler
Indicated Power
Piston Driving Mech.
Net Shaft Power
2
4
6
8
10
Minimum Cycle Press. (MPa)

Fig. 4

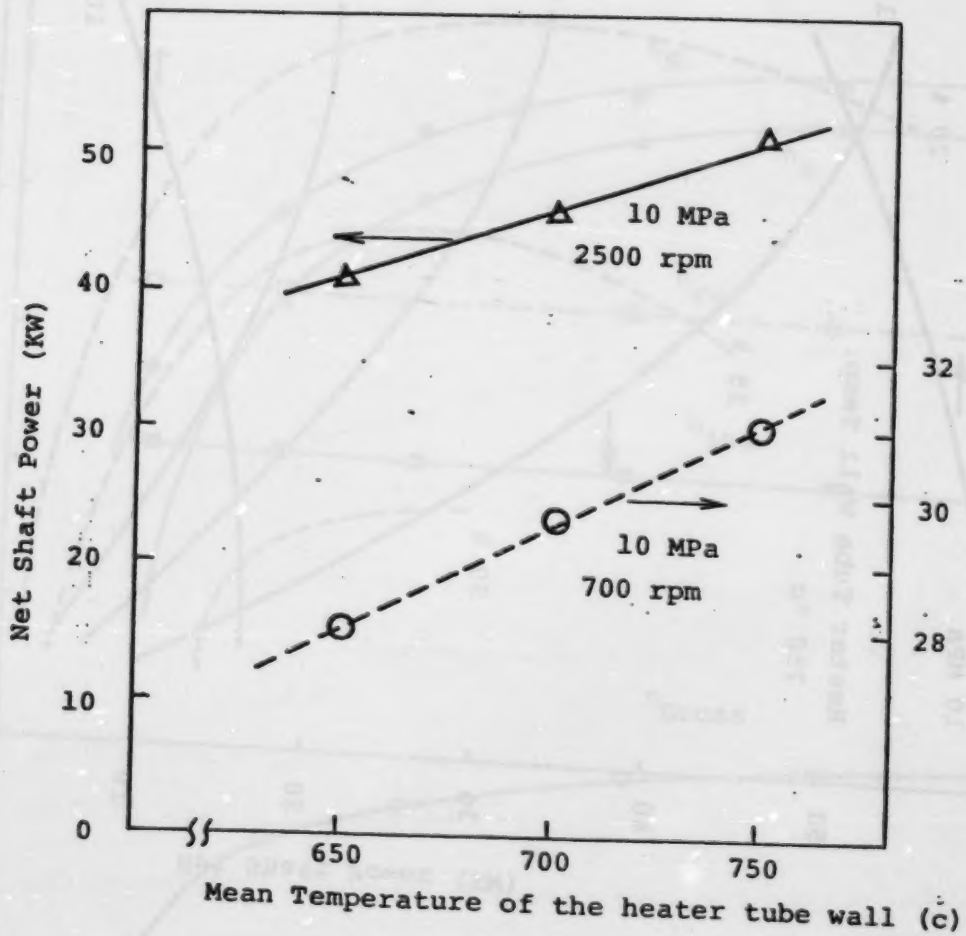

50
40
30
20
10
0
Net Shaft Power (KW)
10 MPa
2500 rpm
10 MPa
700 rpm
32
30
28
Efficiency (%)
650
700
750
Mean Temperature of the heater tube wall (c)

Fig. 5

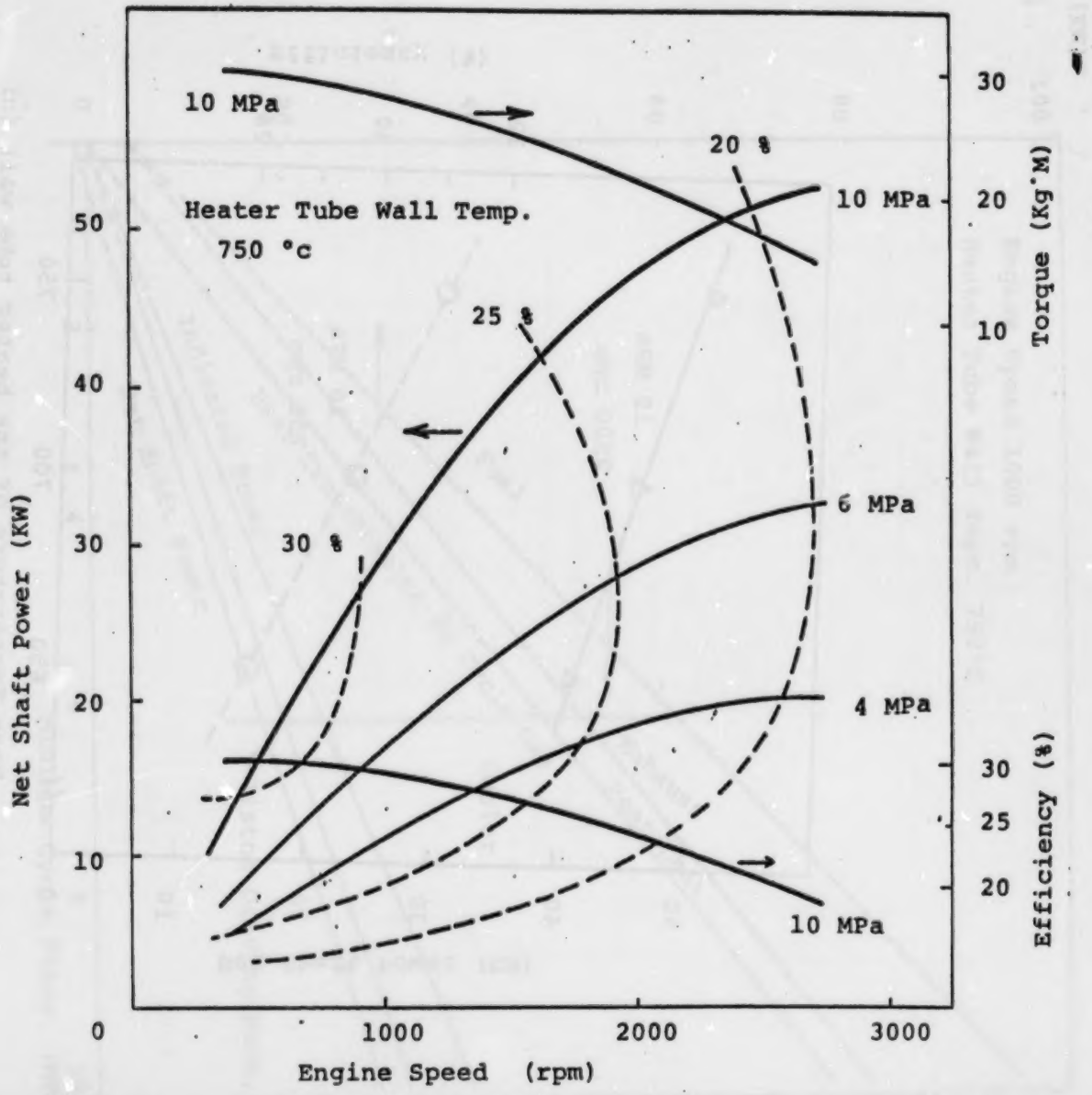
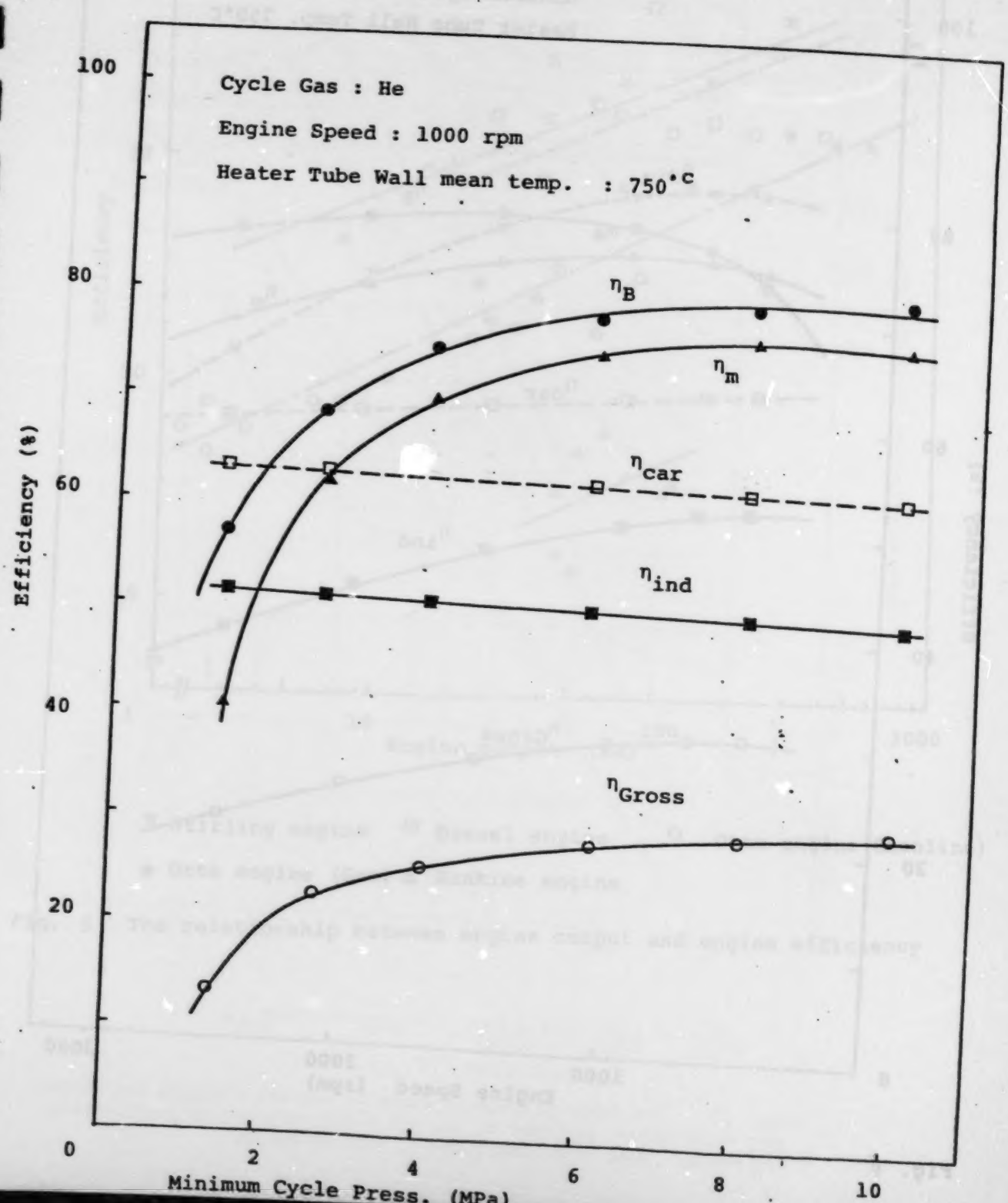

10 MPa
20 %
10 MPa
Heater Tube Wall Temp.
750 °c
25 %
30 %
6 MPa
4 MPa
10 MPa
Net Shaft Power (KW)
Torque (Kg·M)
Efficiency (%)
Engine Speed (rpm)
0
1000
2000
3000
10
20
30
40
50
10
20
30
20
25
30

Fig. 6

Fig. 7


Cycle Gas : He
Engine Speed : 1000 rpm
Heater Tube Wall mean temp. : 750°C
η_B
η_m
η_{car}
η_{ind}
η_{Gross}
Efficiency (%)
100
80
60
40
20
0
2
4
6
8
10
Minimum Cycle Press. (MPa)

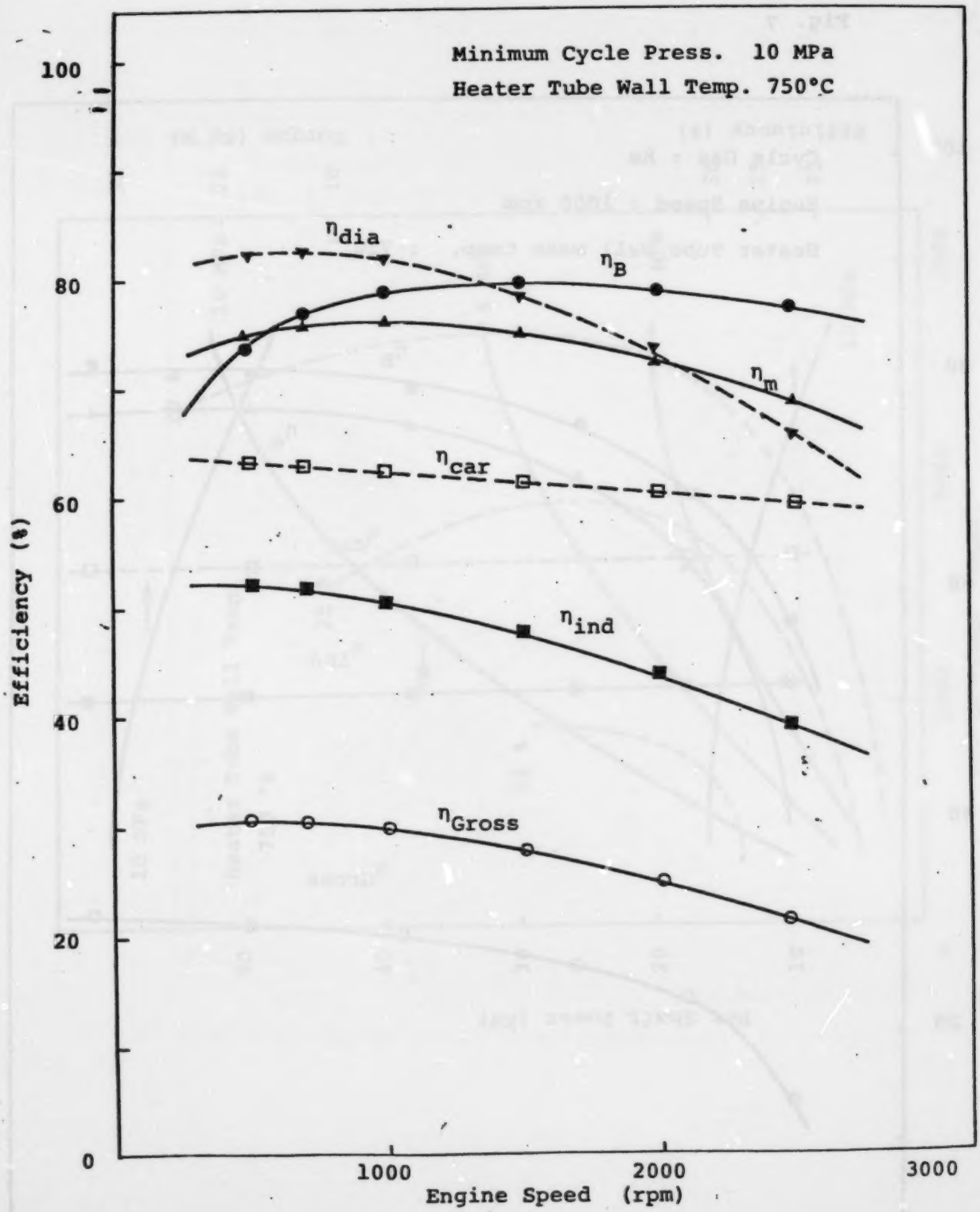
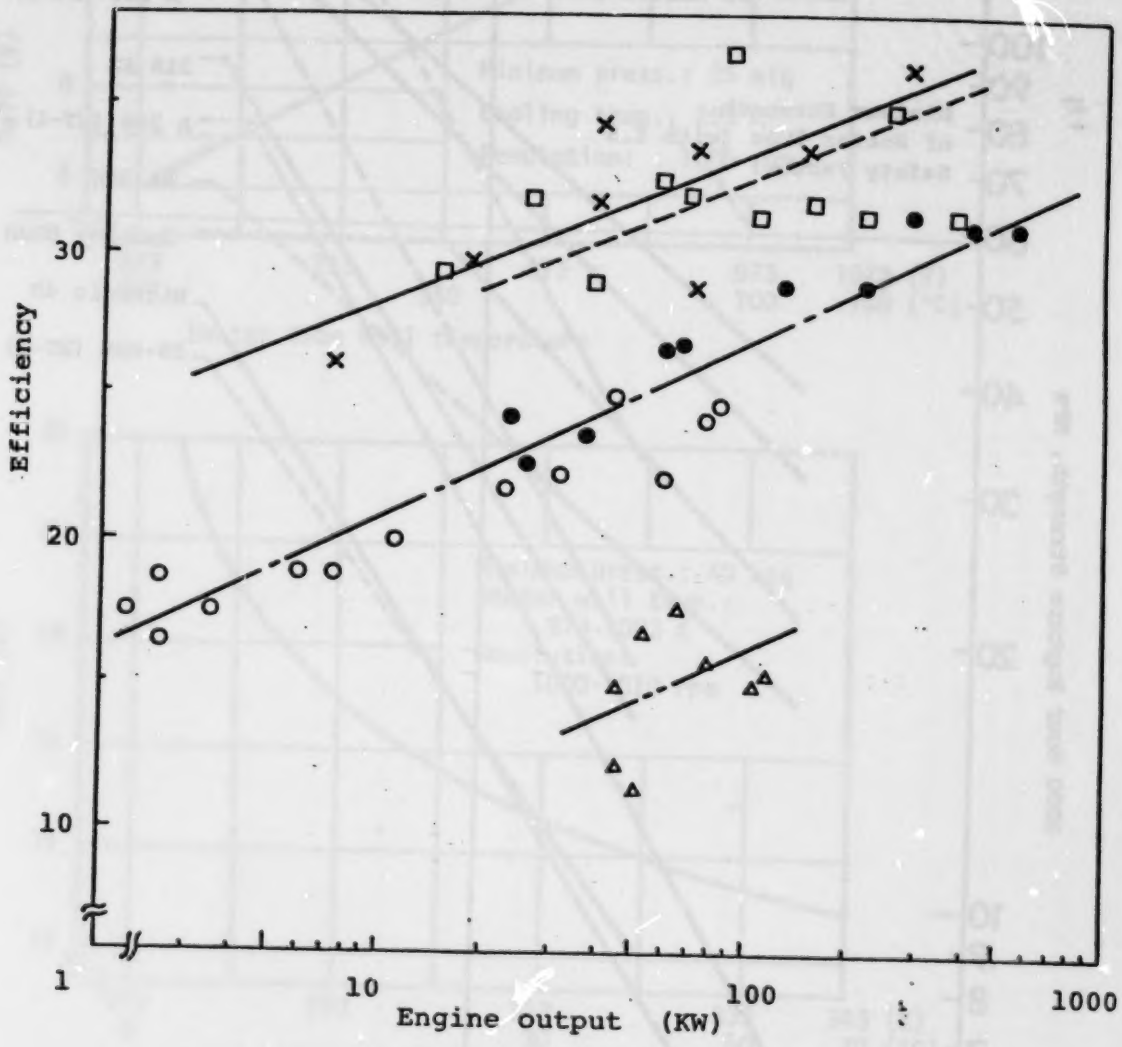

Minimum Cycle Press. 10 MPa
Heater Tube Wall Temp. 750°C
η_{dia}
η_B
η_m
η_{car}
η_{ind}
η_{Gross}
Efficiency (%)
100
80
60
40
20
0
1000
2000
3000
Engine Speed (rpm)

Fig. 8



× Stirling engine □ Diesel engine ○ Otto engine(Gasoline)

● Otto engine (Gas) △ Rankine engine

Fig. 9 The relationship between engine output and engine efficiency

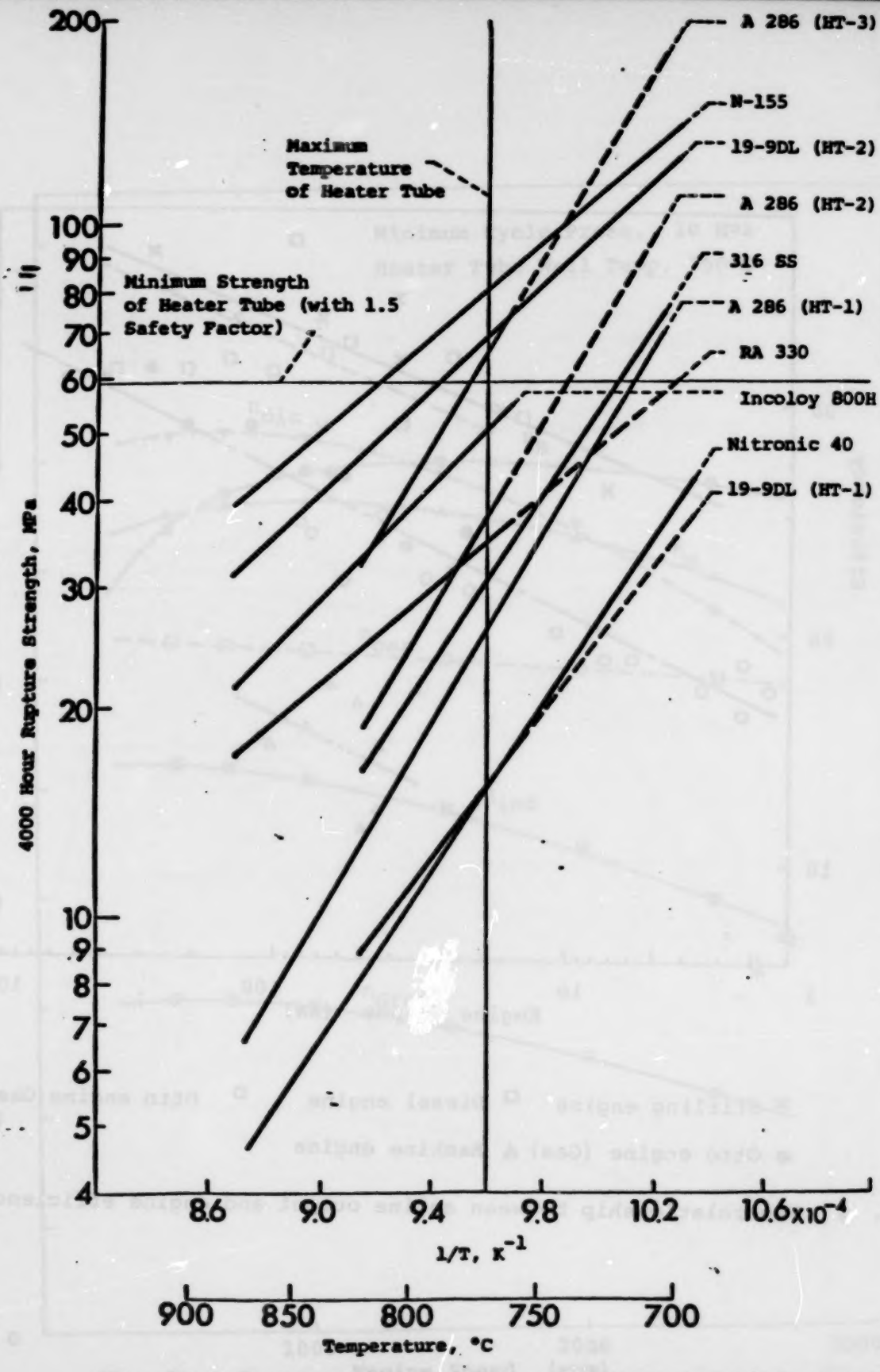
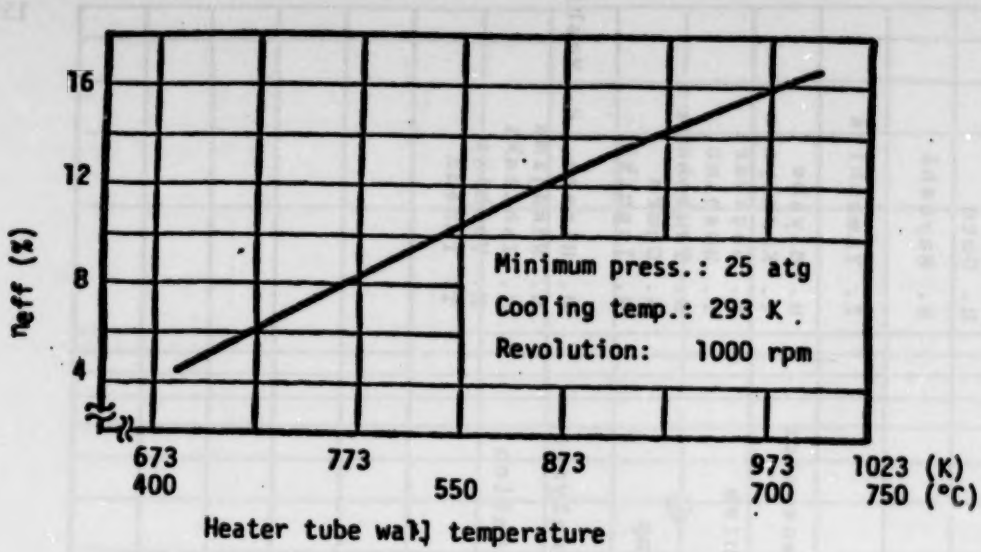


Fig. 10 The temperature dependency of rupture strength for unaged iron-base alloys compared to the Mod I Stirling engine requirements for heater tubes, (DOE/NASA/1040-15, NASA TM-81534)


16
12
8
4
ηeff (%)
Minimum press.: 25 atg
Cooling temp.: 293 K
Revolution: 1000 rpm
673
400
773
550
873
973
700
1023 (K)
750 (°C)
Heater tube wall temperature

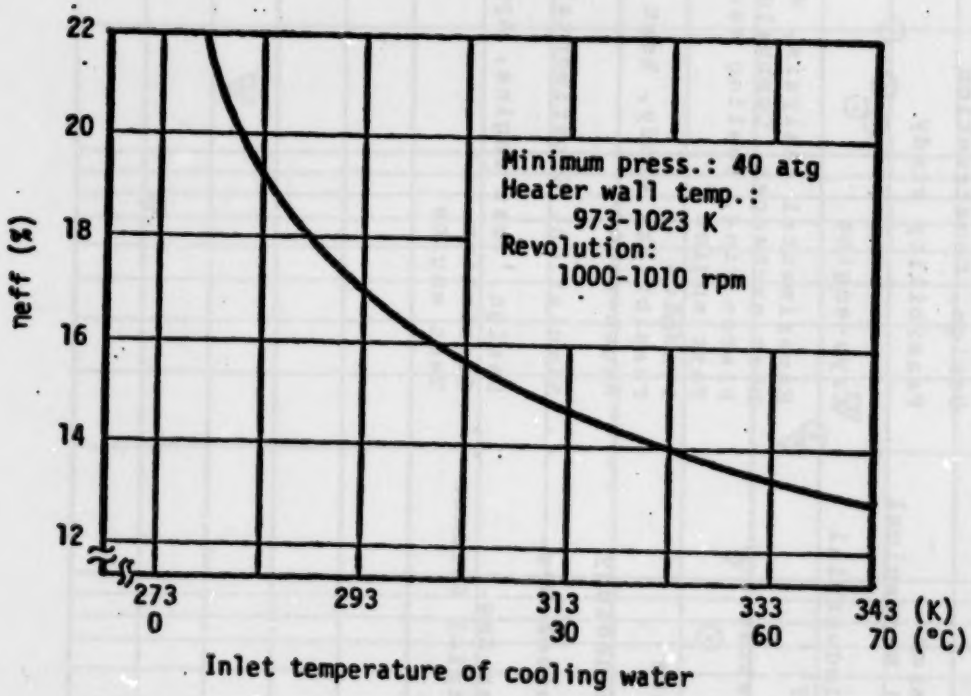

22
20
18
16
14
12
ηeff (%)
Minimum press.: 40 atg
Heater wall temp.:
973-1023 K
Revolution:
1000-1010 rpm
273
0
293
313
30
333
60
343 (K)
70 (°C)
Inlet temperature of cooling water

Fig. 11

Table 1 Research groups

INSTITUTE	RESEARCH BEING CONDUCTED	LEADER
AISIN SEIKI Co.	Development	T. Tani
Daihatsu Diesel	Design, Construction	H. Goto
Japan Automobile Research Institute, Inc. (JARI) Mechanical Engineering	Feasibility study	H. Hayashi
Institute (Agency of Industrial Science and Technology)	Cryo-engine	I. Yamashita
Meiji University	Experimental analysis, Regenerator	H. Miyabe
Mitsubishi Heavy Industries	Heat exchanger, Combustor	T. Kushiyama
Nihon Piston Ring Co.	Piston ring, Sealing mechanism	E. Sugawara
Nissan Motor Co.	Test engine	Y. Hoshino
Ship Research Institute	Dynamics	S. Tsukahara
Tokyo Gas Co.	Feasibility study, heat pump	M. Ogura
Tokyo Institute of Technology	Regenerator	N. Isshiki
University of Tokyo		
Dept. of Mechanical Eng.	Simulation, Experimental analysis	M. Hirata, H. Yoshiki
		N. Nakajima
Dept. of Nuclear Eng.	Design, Test engine, Application	Y. Ishizaki
Inst. of Industrial Science	Simulation	H. Yoshiki
Yanmar Diesel Co.	Test engine	T. Yamada

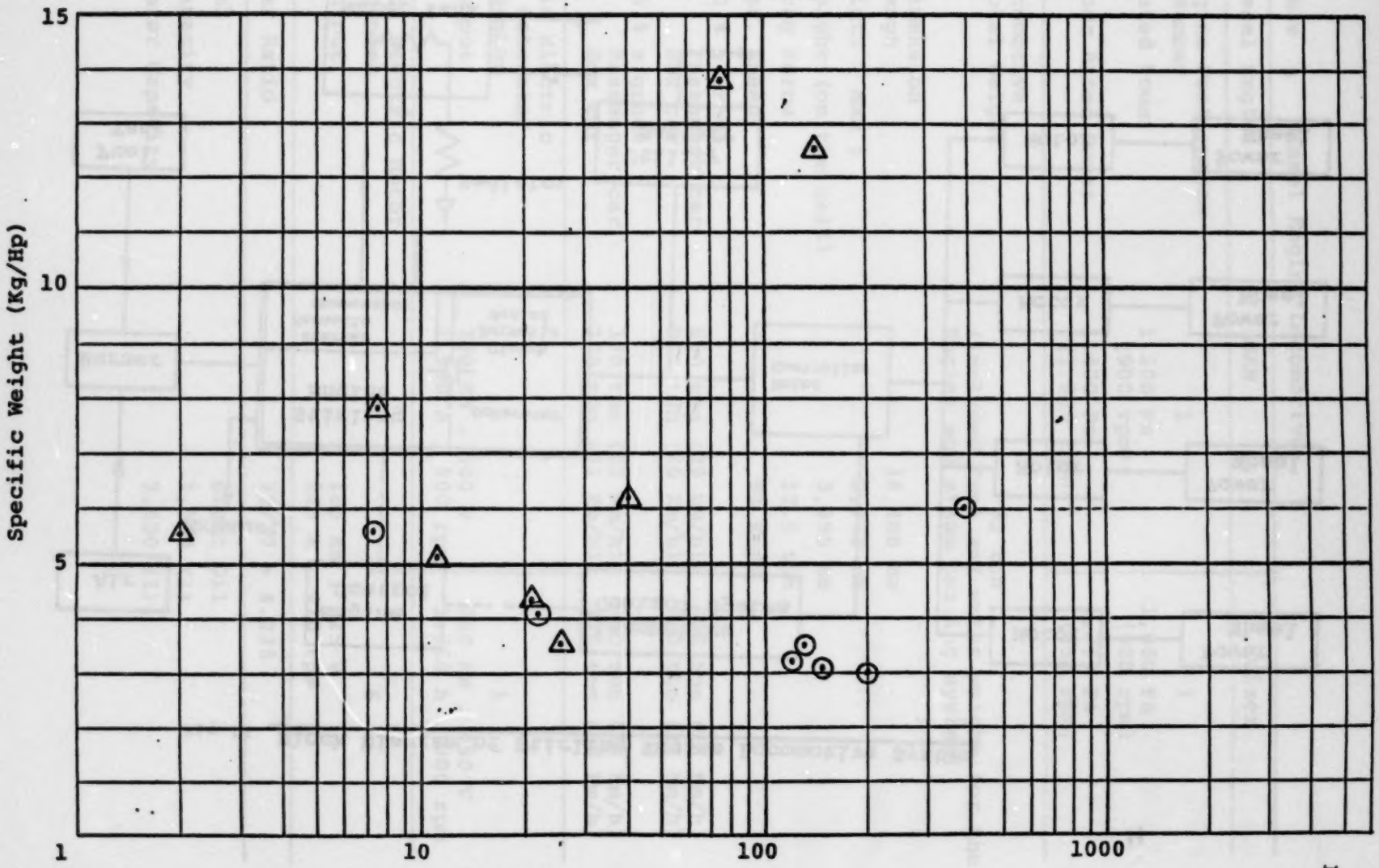


Fig. 12 Stirling Engine Output VS Specific Weight

Fig.13 **Block Diagram of Stirling Engine Locomotive System**

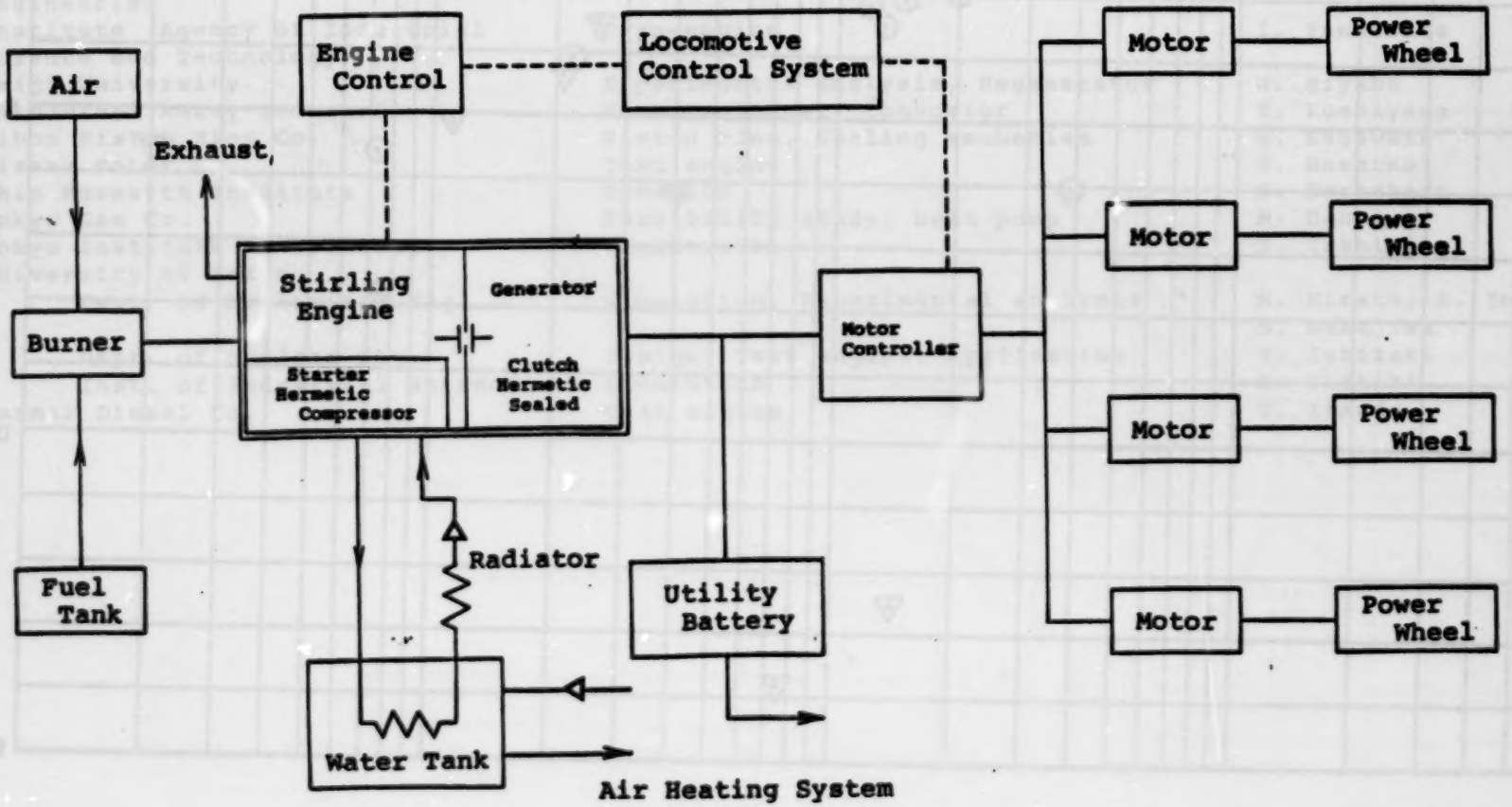


Table 3 Giesel Engine Locomotive

Giesel Engine	MAN	Sulzer
Engine Performance Number	1	1
Rated Power	1,200 Ps (900 rpm)	1,060 Ps (800 rpm)
Hour Rated Power	1,400 Ps (1,000 rpm)	1,200 Ps (850 rpm)
Locomotive Total Weight	81 ton (included Fuel and water weight for the Engine and steam heating system)	
Dimension Lengh	16,400 mm	
Width (Max)	2,932 mm	
Height (on the rail)	3,995 mm	
Drag Force	12.5 ton	
Max. Speed	90 Km/h	
10 % a grade Passenger car	630 ton (29 Km/h)	630 ton (26 Km/h)
Box car	930 ton (20 Km/h)	930 ton (18 Km/h)
25 % a grade Passenger car	330 ton (22 Km/h)	330 ton (20 Km/h)
Box car	370 ton (21 Km/h)	370 ton (18 Km/h)
Main Electric Generator Number	1	1
Output	780 KW , 500 V 1,560 A , 900 rpm	700 KW , 450 V 1,560 A , 800 rpm
Main Electric Motor Number	6	6
Output	100 KW , 225 V 520 A , 410 rpm	
Gear Ratio	72/17 = 4.235	
Fuel	Light Oil	
Capacity	1,500 (l)	
Water Capacity	3,000 (l)	

Table 2 Specification of 50 KW Stirling Engine

Cycle Gas	He Gas
Fuel	White Kerosine
Piston Type	Double Acting
Cylinder	4
Heater Tube Temperature	750°C
Cooling Water Temperature	20°C
Power Pick up Mechanism	Swash Plate Type
Shaft Seal System	Sliding Seal
Power Control System	Minimum Pressure Control System
Combustion System	Super Sonic Wave Atomizer
Air Preheater	Multi Tube

APPENDIX 2

STIRLING ENGINE DEVELOPMENT IN EUROPE

R.A. Billett
Radan Associates
Bath
England

INTRODUCTION

This summary of Stirling engine activity in Europe was prepared at the request of Dr. G. Walker of Calgary, Alberta, Canada, for inclusion in a study of the use of Stirling engines for railway locomotives.

Work on Stirling engines in Europe and elsewhere has been reviewed by Walker in his new book* and we shall not duplicate that here. Rather we shall review the recent developments since the book was written (about 1978).

First we should make clear that as far as is known there is simply no active work concerned with the use of Stirling engines for locomotives. In the recent past a number of railway research centres have indicated an interest in Stirling engines by attendance at meetings and seminars concerned with the topic but have consistently denied the prospect for active development work in the near future. Their interest in Stirling engines was aroused principally because of the favourable environmental characteristics of the Stirling engine.

PHILIPS STIRLING ENGINES

Until quite recently the recognized centre for European Stirling engine activity was of course the Philips Research Laboratories at Eindhoven, Holland. Work on Stirling engines started there about 1938 and continued uninterruptedly for the next 40 years in cooperation with

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*Stirling Engines - 1980. G. Walker, Oxford University Press.

licensees from about 1958 onwards. This whole programme was described in Walker's book referenced above.

About 1978 the Ford Motor Company in the United States, the contemporary partner of Philips in the DOE/NASA Stirling automotive programme, announced the cessation of further research effort on Stirling engines. This, together with a change in the top management at Philips Research Laboratories resulted in a very substantial contraction of the Philips programme. Effort was restricted to a minor heat pump development and it is understood that even this programme has now been abandoned. The production of Stirling cycle cryocoolers continues at Eindhoven but the scale of research effort applied is said to be much less than in previous years. Many of the senior Stirling engine and cryocooler research programme staff are said to have retired over the past five years.

There were two European licensees of Philips, a) United Stirling of Malmo, Sweden and, b) a consortium of the West German engine companies, MAN/MWM.

UNITED STIRLING ENGINES

United Stirling was owned jointly by the Swedish defence related company FFV and the Swedish shipbuilding conglomerate Kockums. The recent collapse of Kockums and its takeover by the Swedish government means that United Stirling is now wholly government owned and directed.

United Stirling have aggressively pursued the development of Stirling engines for various applications

concentrating their effort on 40 kW and 75 kW engines with a dual 75 kW engine of 150 kW output. They foresee multiple applications of the engine for automotive, mining equipment, marine and stationary power application. They have researched the application of heat pipes, the energizing of Stirling engines by wood-waste, the application of Stirling engines to underwater power systems and for automotive use in cars and trucks.

So far as is known the company has not seriously addressed the question of Stirling engines for the large power range required for locomotive use.

United Stirling do not plan to put Stirling engines in production but rather to act as licensees and consultants to other manufacturers. They are participants in the joint Mechanical Technology Inc./Amercian Motors/United Stirling programme for the automotive Stirling engine in North America funded by the U.S. Department of Energy and managed by the NASA Lewis Research Centre.

FFV STIRLING ENGINES

The Swedish company FFV, part-owners of United Stirling have pursued the development of a small 10 kW, Vee-Stirling engine driving an electric power generator. The machine is marketed in North America through Stirling Power Systems of Ann Arbor, Michigan and was incorporated as the power pack in large Winnebago recreational vehicles. The market for these units has declined with the onset of the energy crisis so the future of the FFV machine is unclear.

It is fully developed for limited production but reliability over the long term in commercial usage has not yet been established.

MAN/MWM STIRLING ENGINES

The other major licensee of Philips was a combined effort of the two West German diesel engine makers MAN/MWM. Walker in his book has discussed the very interesting and active contributions made in the early and mid-1970's by the group following commencement of the license arrangement in 1968. The Germany company were known to be interested in 200 horsepower engines and it is said they have developed prototypes in the 800 to 1000 horsepower range but no details of these machines have been disclosed in the open literature.

The range and extent of their present interests in Stirling engines is unknown.

THE BRITISH SCIENCE RESEARCH COUNCIL CONSORTIUM

Very little research on Stirling engines has been done in Britain. Professor Burstall at Kings College, University of Durham started work about 1946 on a programme that continued until about 1963. Various other minor programmes were also carried out in other universities, notably Reading, Bath and Bradford.

In the late 1970's a more substantial effort, funded by the U.K. Government Science Research Council was initiated. The programme was a joint effort of the Universities of Reading and Bath in conjunction with the

Royal Naval Engineering College, Manadon, with Associated Engineering Ltd. and Petters Ltd. also participating. The programme called for the design and manufacture of prototype Stirling engines of 20 kW output. Two engines have been made and installed at the University of Reading and the Royal Naval Engineering College.

In addition to the SRC programme the Royal Naval Engineering College has developed a major interest and research programme in Stirling engines. A United Stirling P40 engine has been acquired for test purposes and substantial numbers of student projects are regularly in progress at the college.

THE HARWELL TMG

A very significant invention in the late 1960's was made by Mr. Cooke-Yarborough of the U.K. Atomic Energy Research Establishment, Harwell. This was the thermo-mechanical generator (TMG), a special type of Stirling engine. It was intended as the energy conversion system for small electric power supplies of very high reliability using Strontium 90 isotope. The cost of the isotope put the system beyond the interest of potential users so the engine was converted to operate on liquid petroleum gas. It is said to be fully developed and likely to find commercial application in the near future.

UNIVERSITY RESEARCH

The 'energy crisis' of the 1980's heightened interest in energy conversion and usage on a broad front including

the use of Stirling engines for heat pumps, and for power generation with coal, wood-waste and other 'biomass'. Many small projects of this nature are known to be in progress in various university engineering departments in Europe. So far as is known no comprehensive directory of these programmes has been prepared and most are likely of the short-term undergraduate student variety.

INTERNATIONAL CONFERENCE

Relatively few scientific and technical papers on Stirling engines have been published in Europe, most authors apparently preferring to use the annual North American based Inter-Society Energy Conversion Engineering Conference as the vehicle for publication. To remedy this, a two-day European conference on applied aspects of Stirling engines has been arranged by the Institution of Mechanical Engineers, London. The conference will be held in March/April 1982 at the University of Reading, England.

Sept. 1978

Prepared for the U.S. Department of Energy

ORO/5392-01
Dist. Category UC-93

Assistant Secretary for Energy Technology
Division of Power Systems
Washington, D.C. 20545

A Technology Evaluation of the Stirling Engine for Stationary Power Generation in the 500 to 2000 Horsepower Range

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1.0 SUMMARY

1.1 Introduction

The current heightened concern over high energy costs and fuel availability has increased the interest in the Stirling engine as a potential prime mover for commercial, industrial and institutional power generation in the capacity range from approximately 500 to 2000 hp. Compared to its major competitors, the Diesel and gas turbine, the Stirling offers the potential for high efficiency, high reliability and long life, quiet operation, and most important, multi-fuel capability.

To date, the bulk of Stirling engine development effort has been devoted to automotive-type applications. By and large, these development efforts have been aimed at producing light-weight, high-specific power, and competitively priced engines for mobile power applications. In comparison, the stationary power generation market requires engines designed for high efficiency and long life, while accepting the inevitable trade-off towards greater bulk and higher cost. Therefore, before one can determine whether the Stirling engine may be a practical improvement over current prime movers for the aforementioned applications, the applicability of automotive Stirling engine technology to large stationary prime movers must be assessed.

This report documents the results of a study undertaken for the Division of Power Systems, for the purpose of assessing the potential and development status of the Stirling engine, and making recommendations for a possible program to develop 500-2000 hp stationary Stirling engines for commercial introduction by the late 1980's.

The principal tasks of the study were:

1. Assess the technical problems which must be resolved in order to develop commercially competitive and available stationary Stirling engines for multi-fuel operation.

2. Outline a program plan for conducting the development work necessary to advance Stirling engine technology to the point where multi-fuel stationary engines could be introduced commercially by the late 1980's.

The following subordinate tasks were also addressed during the course of the study and results are documented in Section 4.0 - Background and Section 5.0 - Technical Requirements - 500-2000 HP Stirling Engines:

1. Describe principles of operation of Stirling engine and history of developments.

2. Synopsis of recent and current automotive Stirling engine developments.

3. Outline differences in requirements and characteristics between automotive (mobile) and stationary Stirling engines.

4. Compare principal characteristics and advantages of Stirling engines for the stationary power generation applications with those of Diesel and gas turbine engines.

5. Discuss types of Stirling engines that should and should not be considered for the 500-2000 hp stationary engine application.

6. Outline typical design and operating characteristics for the recommended types of stationary Stirling engines.

7. Discuss the use of alternate fuels, particularly solid fuels including coal and coal-derived fuels.

8. Describe the expected characteristics of external heating systems for Stirling engines including both liquid and solid-fuel combustors.

9. Discuss the use of indirect heating systems to improve engine performance and increase flexibility in using solid fuels.

10. Identify the high cost components and where possible suggest approaches for reducing costs.

11. Identify the areas of high technological risk and recommend work needed to reduce these risks.

1.2 Methodology

Although an extensive literature review was conducted at the outset, much of the Stirling technology input to this study came from the background experience of the authors and from discussions with Stirling engine developers. Some of the authors (W. Percival, J. Smith) have been engaged in R&D work on Stirlings for several years. This knowledge was supplemented by fact-finding visits to the two principal Stirling engine developers; N.V. Philips, Eindhoven, Netherlands, and K.B. United Stirling AB, Malmo, Sweden. During these visits extensive discussions were conducted concerning the Stirling's potential for the intended application and the remaining technological risk areas where further work is required. Thus, it is hoped that the results and recommendations of this report will reflect the knowledge and concerns of those having the greatest exposure to Stirling engine technology.

1.3 Results and Conclusions

The principal conclusion from the study is that the Stirling engine indeed offers the potential to become a viable alternative to Diesel and gas turbine engines for stationary power generation in the 500-2000 hp range, primarily because it can be developed for operation on solid and other low grade fuels that cannot be utilized in Diesels and gas turbines. Although its multi-fuel capability is the principal attribute, the Stirling engine also offers significant advantages due to its low noise and emissions characteristics. Engine durability and maintenance characteristics are expected to be favorable even though there are some areas requiring further development before obtaining desired engine life. Stirling engine cost is expected to be somewhat greater than a corresponding Diesel, but it is believed that its ability to operate efficiently on low grade fuels will offset this cost penalty and create a viable market for the Stirling.

Detailed results of the various study tasks outlined above are presented in the corresponding sections of the main report and can be located easily by reference to the table of contents. The major conclusions reached during the course of the study are summarized below. These conclusions are segregated into five classifications as follows: General, Combustion System, Engine Design, Future Technology and Technological Risk Areas.

1.3.1 Conclusions - General

1. Stationary Stirling engines can operate with an efficiency essentially equal to that of Diesel engines of comparable size while producing lower levels of exhaust emissions and noise.

2. Stirling engines can be fired with solid fuels or other low grade fuels, such as residual oil, crop-residues and waste materials with development of suitable combustion systems and heat transport systems.

3. Stirling engines can readily be fired with low Btu gas because the high pre-heat temperature makes it possible to achieve stable combustion with reasonable furnace efficency.

4. In production, stationary Stirling engines burning distillate fuels are likely to be 25-50% more expensive than Diesel engines of similar size.

5. Stirling engines operating on solid fuels (coal) are likely to be 30-50% more expensive than Stirling engines running on distillate fuels.

1.3.2 Conclusions - Combustion System

6. When burning distillate fuels, Stirling engines emit lower levels of exhaust emissions than either Diesel engines or gas turbines.

7. Stirling engines can readily burn coal or other solid fuels - the most likely methods are believed to be stoker and fluidized bed combustion systems, although significant development effort is required.

8. Stoker-fired systems would probably need to use preprocessed coal to limit SO_x emissions, because SO_x stack gas cleanup equipment would be too expensive for single engine installations.

9. When burning solid fuels in a Stirling engine it is most likely that indirect heating means will be employed, i.e., an intermediate heat transport system to transfer heat of combustion to the Stirling engine heater head.

10. Effective indirect heating can be achieved by either a liquid metal heat-pipe system or by a pressurized helium or hydrogen convective heat transport loop.

1.3.3 Conclusions - Engine Design

11. The multi-cylinder double-acting engine configuration is believed to be the most suitable type for large stationary engines from an overall cost-effectiveness point of view.

12. 125 hp per cylinder is an ideal cylinder size for a line of engines to cover the 500-2000 hp range because a basic 4-cylinder engine module would provide 500 hp, and 1000, 1500 and 2000 hp units could be obtained by incorporating 2, 3 and 4 basic engine modules to obtain 8, 12 and 16 cylinder engines. Commonality of parts would help keep engine production costs down.

13. There is little risk in scaling up current automotive size double-acting engine designs to the 125 hp cylinder size because:

    a. Displacer-type Stirling engines have been designed as large as 400 hp per cylinder and built/tested as large as 90 hp per cylinder.

    b. Double-acting engines have been built and tested as large as 40 hp per cylinder - but the scale-up to 125 hp involves little risk since scaling rules developed for displacer engines apply equally well to the double-acting engine.

14. Stirling engines for stationary power are most likely to use helium as the working gas, particulary if indirect heating is employed.

15. In order to design an optimum Stirling engine, it is essential to have access to validated analytical design methods (e.g., computer routines). To the best of our knowledge, the only such analytical routines are available only through Philips and its licensees.

16. Near-term Stirling engines (late 1980's) will use metallic heater heads operating at temperatures ranging from 750-800°C.

1.3.4 Conclusions - Future Technology

17. Future developments in ceramics may offer the possibility of lower cost ceramic heater heads operating at higher temperatures (up to 1100°C) for improved efficiency and power.

18. While ceramic heater tubes may be attractive for use in fluidized bed combustion systems from a cost and corrosion resistance view-point, their heater temperatures will be restricted to below 1800°F to retain effectiveness of sulfur removal and/or to prevent ash fusion.

1.3.5 Conclusions - Technological Risk

The principal areas of technological risk are:

1. Solid Fuel Combustion Systems.

2. Indirect Heat Transport Systems.

3. Heater Head - Cost and Life.

4. Piston Rod Sealing.

5. Piston Seals.

6. Air Pre-Heater - Cost and Life.

While it is not an area of technological risk, regenerator cost reduction may also be required to make the large Stirling engine economically viable. These risk areas are summarized briefly in Section 2.0 (Technological Risk Areas) and discussed in more detail throughout Section 5.0 (Technical Requirements).

1.4 Recommendations

It is recommended that a program be undertaken by DOE to develop the technology for a large stationary Stirling engine capable of operation on solid fuels for possible commercial introduction by the late 1980's. This program should have two principal branches:

1. Large Stirling Engine Development - results would be applicable for engines operating on liquid fuels, as well as solid fuels and other heat sources.

2. Combustion System Development - should provide acceptable methods for utilizing low grade fuels, such as coal and coal derivatives, cultivated fuels, agricultural and forest waste, etc.

Further details of the recommended development program are contained in Section 3.0 (Program Plan).

5.2.2 Engine Scaling Rules

Specific analytical studies of size effect for displacer engines were made by Philips and other licensees during the 1960's. The results became known as "scaling rules". They considered how scaling geometrically similar rhombic drive engine designs, covering a power range from about 10 bhp to over 3000 bhp per cylinder, would affect power, efficiency, stresses, heat transfer, sealing, etc. The studies are analagous to the Massachusetts Institute of Technology, Sloan Laboratory study made in the 1940 - 50's on geometrically similar internal combustion engines (Ref. 39). The scaling rules generally apply to other Stirling engine designs, such as the double-acting type.

Stirling engine scaling rules are like those for internal combustion engines in many respects for geometric similarity, and based on optimizing one size near the middle of the size range. Mean gas pressure, cooling water temperature and average heater tube temperature are assumed constant. When the major linear dimensions of the engine are scaled by the scaling factor, γ, the results are:

1. Bhp is proportional to the scaling factor (γ) squared, to a first approximation.
2. Engine speed is inversely proportional to the scaling factor; i.e., piston speed is held constant.
3. Stresses due to gas pressure and inertia forces remain constant; thermal stresses will increase with γ, but not linearly.
4. Efficiency remains nearly constant over a power range of about 50 to 1.

5. Heat transfer components, optimized for maximum engine performance do not scale according to exact geometric similarity, but overall thermal performance remains nearly constant.

For example, heater tube spacing does not scale as rapidly as γ, but more as $\gamma^{1/3}$*. Heater tube diameter tends to scale as γ, but the hydraulic diameter in the regenerator and cooler do not scale as rapidly as γ.

Applying these scaling rules to existing engine designs (e.g., Table 4-1), the piston diameter for 125 bhp cylinder will be about 7", and the piston will weigh approximately 35 lbs. From consideration of manufacturing, assembly and maintenance, the size and weight are within the limitations for handling by one man. Piston stroke is 4", and engine speed at maximum power approximately 1200 rpm. This speed is suitable for direct driving a 6 pole generator.

5.2.3 Typical Specifications for 500 - 2000 HP Stationary Stirling Engines

Although stationary Stirling engines may take on different forms and have different characteristics, depending on specific application, this section will describe the specifications and characteristics of a "representative" stationary engine. It is assumed that the primary function of the engine is electric power generation, so it will operate at constant speed with load varying according to demand. The utilization factor is expected to be high so the engine design should favor high efficiency and long life at some sacrifice of specific power (hp/ft^3 or hp/lb).

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* See Appendix D for detailed explanation.

Some derating should also be done in choosing the full load operating point to favor high efficiency and durability.

Since the primary advantage of the Stirling engine is its ability to burn solid fuels and other low grade fuels which would probably require indirect heating means (see Section 5.3.2), the "representative" stationary engine will be based on a sodium heat-pipe system to transfer combustion heat to the heater tubes (section 5.3.2.1). When the heat-pipe system is employed, current technology permits operation with heater tube temperatures of 800°C (1470°F) (1% creep in 10,000 hours), so this temperature is assumed for the "representative" engines. Cooling water temperature is assumed to be 30°C (86°F) which is typical for systems employing a wet cooling tower or equivalent.

Section 5.1.2 set forth the logic for choosing a series of multi-cylinder double-acting engines with 125 bhp per cylinder to cover the 500 - 2000 hp range. Thus, engines of 4 cylinder, 6 cylinder, 8 cylinder, 12 cylinder and 16 cylinder (all employing the same basic cylinder size and internal components) would have outputs of 500, 750, 1000, 1500, and 2000 hp respectively. The 4, 8, 12 and 16 cylinder engines would be built up from multiples of 4-cylinder double-acting engine power units similar to the United Stirling P-75 engine (Figures 5-4, 4-21, and Ref. 37). The 6 cylinder version could be implemented with a single 6 cylinder double-acting power unit.

The preliminary design of the "representative" engine series was accomplished by scaling from the most recent United Stirling heavy-duty truck engine, the P-75 unit (Ref. 37). This is a 4-cylinder double-acting "U" type twin crank engine (Figure 5-4) with a 86 mm bore, 48 mm stroke and 100 hp (75 kW) output, when operating direct-fired with hydrogen working gas, 720°C heater temperature and 70°C cooling water. The actual engine

:aling was done by starting with a proprietary United Stirling engine ap for the P-75 engine optimized for a heat-pipe heater head with helium working gas, and 800°C heater tube temperature and 30°C cooling water temperature (approximately 27°C in and 33°C out). At the derated speed of 1800 rpm and a mean working gas pressure of 14 MPa (2030 psi) the power output is 111 hp (83 kW) or 27.75 hp/cylinder. Since the power output varies as the scale factor γ squared, then to scale up to 125 hp/cylinder the scale factor becomes:

$$\gamma = \sqrt{125/27.75} = 2.12$$

Then the bore, stroke and rpm scale as follows:

Bore = 2.12 x 86 mm = 182 mm (7.18 in)

Stroke = 2.12 x 48 mm = 102 mm (4.0 in)

RPM = 1800 ÷ 2.12 = 850 RPM

Table 5-1 summarizes these foregoing specifications for a 2000 hp, 16 cylinder engine. The size is estimated by scaling up from the P-75 engine as the basic 4-cylinder power unit using the scaling factor of 2.12. Overall dimensions are estimated to be 143 in (3632 mm) long, 50 in (1270 mm) wide and 68 in (1727 mm) high. Weight is estimated at 32,000 lb (14,550 Kg), which represents a specific weight of 16 lb/hp. This is almost identical to the specific size and weight of equivalent power Diesel engines used for similar service, e.g., Alco Model 8V251G, (Table 4-2).

The 16 cylinder engine consists of four, 4-cylinder double-acting power units arranged in-line with twin crankshafts carrying 8 cylinders each. A single combustion system operating on either liquid or

solid fuel is mounted on one side of the engine where it supplies heat to the heater head assemblies of the four power units through a sodium heat-pipe system. Since in the twin crankshaft arrangement, the output shaft is geared from the two cranks, any desired output shaft speed could be employed, e.g., 1200 rpm to direct drive a 6 pole generator.

5.2.4 Estimated Performance Characteristics

An engine power and efficiency map for the United Stirling P-75 truck engine is shown in Figure 5-9, (Ref. 66). This is a direct-fired hydrogen engine. Power increases as the mean working gas pressure and operating speed are increased. The maximum efficiency of 37% occurs at about 1/3 speed and at maximum gas pressure.

The "representative" stationary engine discussed in the previous section was scaled from a United Stirling proprietary engine map for the P-75 engine optimized for a heat-pipe heater head and helium gas as discussed above. The engine performance map for this representative engine is shown in Figure 5-10. It was constructed by scaling from this proprietary map and making appropriate allowances for power consumed by the various engine auxiliaries such as combustion blower, oil and water pumps, fuel pump and atomizing air pump. Among these components the combustion air blower is the most significant, consuming an average of 3-1/2% of engine power. All other auxiliaries combined only consume about 1-1/2% of engine output. The performance map does not include any allowance for a cooling water radiator or cooling tower fan or a helium compressor. If an installation employing air cooling is contemplated, then an appropriate allowance would need to be made for the fan. Although the helium compressor power was not included, it will be very small because it is only running a small fraction of the time.

Note that the maximum thermal efficiency of this engine is about 40%, and also occurs at about 1/2 speed and maximum pressure. The operating point for the stationary power generation application has been selected at 850 rpm (about 70% speed) and 14 MPa cycle pressure to provide

DOE/ET/15209-T1
Distribution Categories UC-95 and UC-95f

LARGE STATIONARY STIRLING ENGINE

DESIGN AND DEVELOPMENT OF STIRLING ENGINES FOR STATIONARY POWER GENERATION APPLICATIONS IN THE 500 TO 3000 HP RANGE

CONTRACT NO. DE-AC02-79ET15209

SUBTASK 1A REPORT

MARCH 1, 1980

STATE-OF-THE-ART CONCEPTUAL DESIGN

PREPARED FOR:

U.S. DEPARTMENT OF ENERGY

WASHINGTON, D.C.

GENERAL ELECTRIC

ADVANCED ENERGY DEPT.

SPACE DIVISION

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1.0 SUMMARY AND OVERVIEW

This report summarizes the first portion of the Conceptual Design of Stirling Engines for Stationary Power Application in the 500 to 3000 HP range which was aimed at state-of-the-art stationary Stirling engines for a 1985 hardware demonstration. (Task 1A)

The main goals of this effort were to obtain reliable cost data for a Stationary Stirling engine capable of meeting future needs for total energy/cogeneration systems and to establish a pragmatic and conservative base design for a first generation hardware.

In order to secure a competitive position for this engine in the marketplace, it was clearly recognized that a coal and waste product fired version had to be given primary consideration since a fuel oil or gas fired version would not be cost competitive with a diesel/gas engine. Therefore, the mating of the Stirling engine with an Atmospheric Fluidized Bed (AFB) was a main target.

Since cost effectiveness (or ROI) for total energy/cogeneration systems depends - besides the initial capital investment - mainly on the total operating time per year (high capacity factor), a main objective of the design was not only low cost, but high reliability and good maintainability with engine efficiency being an important but not main consideration.

Starting with an extensive screening effort, four (4) engine types and three (3) heat transport systems were selected:

ENGINE TYPES	HEAT TRANSPORT SYSTEMS
• V-type crank engine	• Heat pipe
• Radial Engine	• Pressurized gas heat transport loop
• Swashplate engine	• Direct gas fired system
• Rhombic drive engine	

After a preliminary layout cycle, the rhombic drive engine was eliminated due to intolerable maintenance difficulties on the push rod seals. "V", Radial and Swashplate engines were taken through a detailed design/layout cycle, to establish all important design features and reliable engine weights, including all auxiliaries.

A costing program - RCA's "PRICE 84" - was employed to establish engine costs (including all engineering development and testing costs) as a function of engine size and yearly production rate ranging from 10 to 10,000.

A summary of this weight and cost analysis is given in Table 3.3-11. Specific weights range from 9.4 lb/hp (radial) to 12.5 lb/hp (swash-plate) and specific costs were obtained between 230 $/hp* (308 $/KW) and 297 $/hp* (398 $/KW) for a 1000 HP engine at a production rate of 1000 engines per year. (Comparative data for a 2000 HP locomotive diesel engine: 13.5 lb/hp and 117 $/hp or 157 $/KW).

It has to be noted that a quite conservative approach was deliberately chosen for the first generation hardware:

- Helium instead of hydrogen as working fluid - mainly for market acceptability.
- 100 atmoshperes mean pressure level instead of 200 ATM now being considered in automotive Stirling engines.
- 1200 RPM engine speed - with a good change for later upgrading to 1800 RPM.
- Bearings and crankshafts consistent with high peak load diesel design practices rather than higher allowables for the more even load distribution on Stirling engines.

After comparing engine layouts and analyzing qualitative and quantitative evaluation criteria, the V-crank engine (Figure 1-1) was chosen as the candidate for a 1985 hardware demonstration. This choice is substantiated by the following advantages:

- State-of-the-art drive with considerable experience and established reliability from large diesel engines for the entire drive train and auxiliary system.
- Existing production facilities (forging, heat treating, machining, grinding and balancing).
- Existing skills for design, assembly and maintenance, no development required for the drive system, crankcase, auxiliaries.
- Wide application range and interface flexibility.
- Good modularity.
- Good component accessibility for easy maintenance.
- Enhanced market acceptability due to familiarity with components and configuration.

The radial engine design with four cylinders (90°) in one plane is ideally suited for a Rinia type engine, has good modularity and excellent weight and cost characteristics (approximately 20% lower than "V" engines). This engine was not chosen since maintainability

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* Including all auxiliaries but not the Heat Transport Loop.

considerations make a vertical shaft mandatory which curtails application flexibility and reduces an already lower market acceptance. However, this engine might be a good supplement to an extended production program for second or third generation hardware.

The swashplate engine, even though compact in design, did not exhibit the same weight and cost advantages as the radial engine with considerable reduction in drive system maintainability (limited access to critical bearings). Furthermore, the development of a large swashplate drive would introduce a considerable risk to program cost and schedule.

The proposed "V" engine design incorporates the following novel and unique features:

- Individually mounted water cooled cylinders with integrated seal cartridge for easy exchange and maintenance.
- Seal system without catastrophic failure modes which can tolerate considerable leakage to minimize forced outages.
- Exchangeable heat exchangers, regenerators and coolers with low temperature, non-welded interface joints for easy maintenance and heat source adaptability.
- Interchangeable heat transport systems for fuel flexibility and later fuel conversion capability.
- Extensive use of low cost material through novel design approaches for cylinder, regenerator/cooler and heat transport loop manifold design.
- Novel dead volume control system.
- Integrated single bearing generator for ease of site installation.
- Easy adaptability for advanced hardware and retrofit applications.

The chosen design represents a conservative approach with a high degree of confidence for the viability of a demonstration and first generation hardware.

In an evolutionary process, later generation hardware may be upgraded to more than twice the power level/per cylinder with minor modification in engine hardware by introducing higher pressure levels, 1800 RPM engine speed and hydrogen as working fluid.

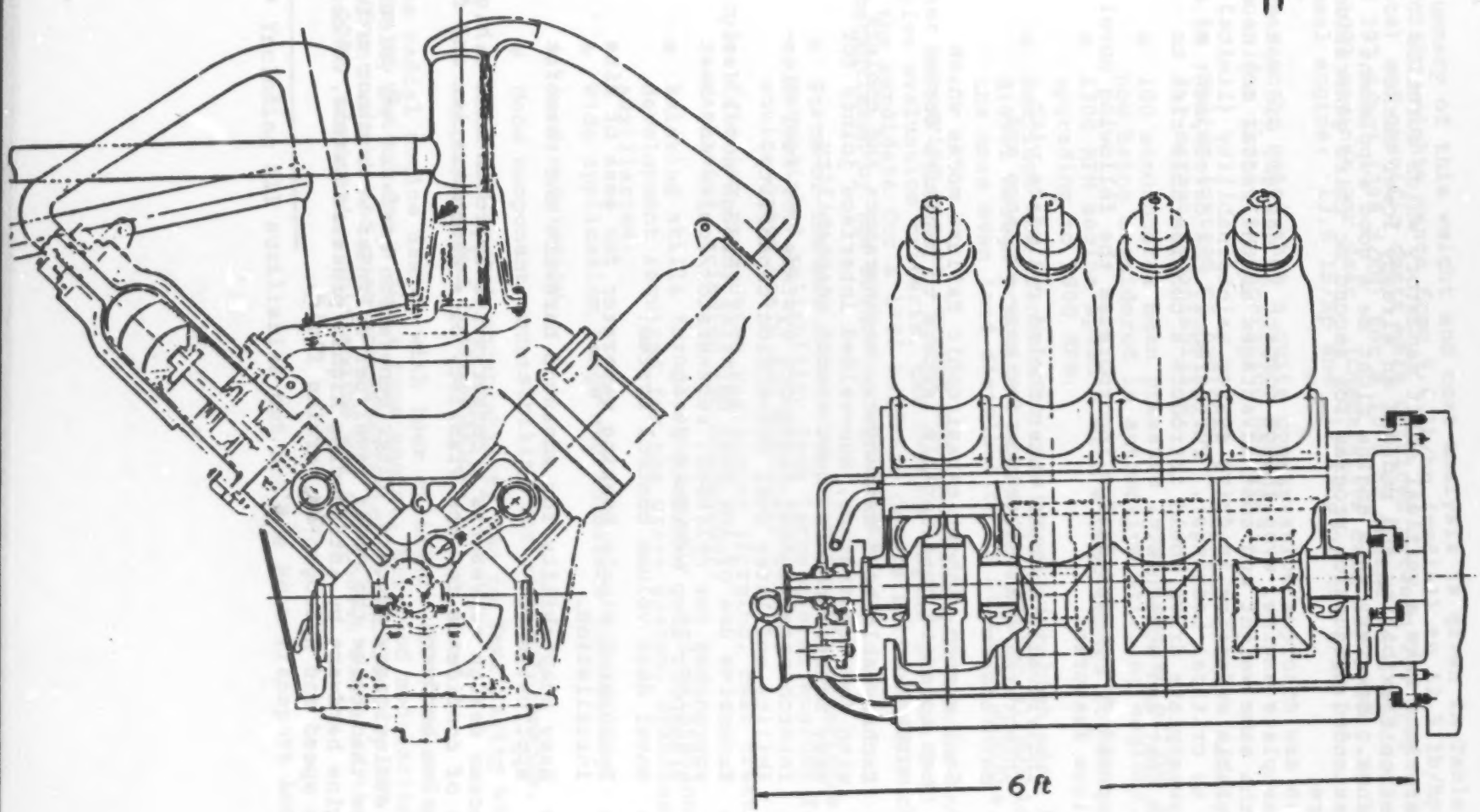

Figure 1-1. Selected State-of-the-Art 1985 Demo Engine 1000 HP-V8

For a successful engine demonstration by 1985, the following areas require further attention:

- Reliable seals for this size of engine are not state-of-the-art and further design and development testing is needed before a final design can be chosen.
- Reliable heat source cost data are not available making an economic assessment difficult. Economically viable designs for the total energy/cogeneration market have to be established which fall in the desired output range and address problems such as modularity, turn-down ratio and coal handling equipment for small outputs.
- A heat transport loop is a necessary component to achieve good fuel and system flexibility.

Heat pipes of the required size (250 KW thermal) are not state-of-the-art and exhibit certain operational problems which are not sufficiently resolved.

A pressurized gas heat transport loop suffers from the non-availabilit of suitable high temperature gas circulators at a pressure level of approximately 40 atmospheres. Furthermore, such a system adds approximately 60 $/hp to the overall engine cost (vs. 25 $/hp for the heat pipe system) and has an auxiliary power consumption of approximately 5% of engine shaft power.

A considerable design and development effort is required to close the gap to a truly state-of-the-art heat transport system equipment.

The degree of detail in which this conceptual design was carried out was only possible with the close cooperation and help from many experienced contributors:

- GE-Locomotive Operation contributed many design details taken from existing locomotive diesel hardware, assisted in many valuable reviews and provided all team members a first hand familiarization with similar type hardware, including assembly and test operations.
- GE-Energy Technology operation in Schenectady, NY is responsible for the economic analysis using an existing computer program modified for the specific needs of this program.
- Foster Wheeler Development Corporation contributed a valuable heat source selection study and interface definition as well as conceptual layout for a typical AFB heat source.
- Dr. R. Meijer and B. Ziph - N.V. Philips - provided Stirling component design guidance.
- Thermacore, Inc. assisted in heat pipe definition.

A swashplate or "Z" crank drive could incorporate 4 to 6 cylinders on one side of the drive mechanism; also, the same number of cylinders might be placed on the opposite side giving an engine with 8, 10 or 12 cylinders in a "barrel" or axial arrangement. This latter configuration might prove awkward particularly since the central output shaft can intrude into or otherwise complicate the combustion chamber arrangement.

Another possible configuration incorporates multiples of 4 cylinders in a "square" arrangement with long twin crankshafts, resulting in an in-line engine of 8, 12, 16 or even 20 cylinders. (Twenty cylinders in a "V" arrangement is considered the upper limit in Diesel engine practice because of crankshaft and crankcase design problems, in addition to maintenance problems, as the number of cylinders increase.)

An alternative to the above is an "opposed" or perhaps "radial" design with either horizontal or vertical crankshafts (see sketch Figure 5-5). This system has the advantage of permitting 2 cylinders per single crank throw, as in the "V" type Diesel engine. It could be built with 8 cylinders opposed having 2 crankshafts either vertical or horizontal, or a 16 cylinder "X" engine with 4 vertical crankshafts. Another, but perhaps less likely, candidate is a 3 crankshaft "V" probably with the shafts horizontal.

5.1.3 Piston and Rod Seals

Perhaps the most critical mechanical components in the Stirling engine are piston rings and piston rod seals. A great deal of R&D effort has been devoted to these components over the past 20 years and significant progress has resulted. However, there is still need for further improvement to provide the performance and durability needed for stationary engines.

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a. Piston Ring Sealing and Wear

Stirling engine piston rings must prevent excessive gas leakage by the piston or displacer while minimizing mechanical losses due to ring friction. Liquid lubricants cannot be used to improve friction and wear characteristics because any lubricant entering the hot sections would thermally decompose and contaminate the system (especially the regenerator matrix) with solid residue, and dilute the H_2 working gas with high molecular weight hydrocarbons, such as methane (Ref. 33, p 9). Therefore, solid lubricant type rings must be employed and their wear characteristics become the critical factor.

At present, the rings are made of a polymer, PTFE, or Teflon, which is impregnated with fiberglass and sold commercially as Rulon, Fluon, and other trade names. These rings have proven satisfactory for use in displacer type engines where piston ring life in excess of 4000 hours has been demonstrated and ring life as high as 17,000 hours has been projected based on measured wear rates (Ref. 33). However, double-acting engines have more severe piston ring operating conditions so further improvements may be required. The maximum pressure differential across the rings for a double-acting 4 cylinder engine is approximately 40% greater than for a displacer engine, if both are operating at the same mean pressure and assuming the displacer engine is equipped with a buffer space operating at constant mean pressure.

The wear rate for polymer type piston rings is a function of the P-V product (product of mean pressure differential and mean piston velocity). As the pressure differential across the ring increases, the wear rate goes up and there is also a tendency for this type ring to be extruded into the clearance spaces. Temperature also affects ring wear rate, but the relation is more complex and is not well understood at this time. In order to keep ring temperatures as low as possible,

Stirling engines employ long hollow pistons with the sealing rings at the cold end where they are protected from exposure to the hot working gas. Also, the rings ride on a relatively cool area of the cylinder wall, frequently water cooled. As long as the ring leakage is small, the hot leakage gas is cooled nearly to ring belt temperature before passing through the ring. This is borne out by experience which shows that wear rates are nearly the same for both upper and lower rings, implying that the most important factor is the temperature of the surface on which the rings operate.

The pressure differential across the power piston sealing rings changes direction twice during each revolution of the crankshaft. This causes the sealing rings to move from one side of the ring groove to the other and leads to erratic sealing behavior. One of the problems that has resulted from this phenomenon is a tendency for double-acting pistons to pump gas between cylinders resulting in dangerously high pressure in one engine module and "starvation" of gas in others. In order to obtain more consistent ring sealing behavior and solve this "pumping" problem, recent double-acting engines have employed two sealing rings per piston as shown in Figure 5-6. A groove cut into the piston between rings is connected to the internal volume of the piston. By slots cut in the outer sides of each ring, the rings are made to pump outward which tends to produce minimum pressure in the volume between the rings. True minimum only occurs with infinite volume, however, and as a result of finite volume, there is greater leakage between hot and cold spaces across the rings. Nevertheless, this scheme has helped to maintain uniform mean pressure among cylinders while holding ring leakage to acceptable values.

Polymer type piston rings with various fillers including fiberglass are used in commercial air and gas compressor service. For reasons not yet fully explained, the wear rates of these rings are from one to two orders of magnitude greater than similar rings used in Stirling

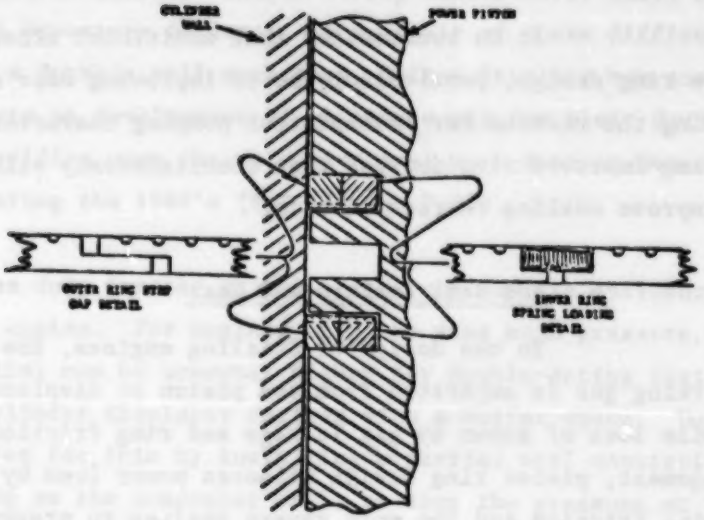

Figure 5-6 Typical Piston Sealing Ring Configuration (Ref. 33)

engines. This includes operating experience over many thousands of hours at Philips, General Motors and United Stirling. The lower wear rates in Stirling engines are at least in part, due to the clean, inert working gas environment (no oxygen present).

It is recommended that additional effort be made on piston ring design, particularly as to improving wear resistance, understanding the reasons for inconsistent pumping characteristics, and developing improved ring designs that simultaneously eliminate pumping and improve sealing characteristics.

b. Rod Seal Leakage and Wear

In the design of Stirling engines, the sealing for loss of working gas is separated from the piston or displacer sealing to minimize loss of power by gas leakage and ring friction. With this arrangement, piston ring design balances power loss by gas leakage versus ring friction and the more severe sealing to prevent loss of working gas to the environment is at the piston rod seal at the cold end of the cylinder. A tighter seal can be achieved without excessive friction, since the seal length is shorter than for a piston seal. Thus, gas leakage past piston rod seals is more critical than piston ring leakage because it represents a loss of working fluid from the system which must either be recovered, cleaned and pumped back into storage, or supplemented by make-up. Besides minimizing gas leakage, the rod seal must also prevent the crankcase lubricating oil from invading the engine working gas space, and fouling the hot heat exchange components and working gas with decomposition products.

Two basic types of rod seals have been used on Stirling engines to date: sliding seals and the "roll sock" seal invented by Philips in 1960 (Ref. 7). Although the "roll sock" seal is attractive

because of its zero leakage characteristic, the developments to date, have not produced sufficient or consistent seal life (Ref. 33, pp 21-28). The rolling diaphragm, usually made from polyurethane material, has proven to be very sensitive to quality control during manufacture, small occluded impurities causing premature failure. As a result of these difficulties in development of a durable roll sock seal, United Stirling has concentrated their efforts on development of sliding seals for their double-acting engines, building upon the extensive seal work done at General Motors Research during the 1960's (Ref. 33, pp 7-20).

The duty imposed on the piston rod seal is influenced by the design of the engine. For engines with the same mean pressure, the pressure differential can be somewhat higher for double-acting designs than for single-cylinder displacer engines with a buffer space. United Stirling compensates for this by installing a partial seal consisting of a modified "O" ring as the innermost seal, to drop the pressure at the main seal to somewhere near the mean pressure level -- the equivalent of the pressure in the buffer space of a displacer engine (Figure 5-7).

Using state-of-the-art sliding rod seals, United Stirling obtains leakage of hydrogen from the 4 piston rod seals of a 100 bhp, 4 cylinder double-acting engine ranging from 0.1 to 0.2 grams/hour at full load. The life of their piston rings, as well as rod seals is currently about 4000 hours. Materials on which the rod seals operate are not as important as is their hardness, roundness, straightness and surface finish. Experience at General Motors is described in Ref. 33. Quoting from that reference, p 19:

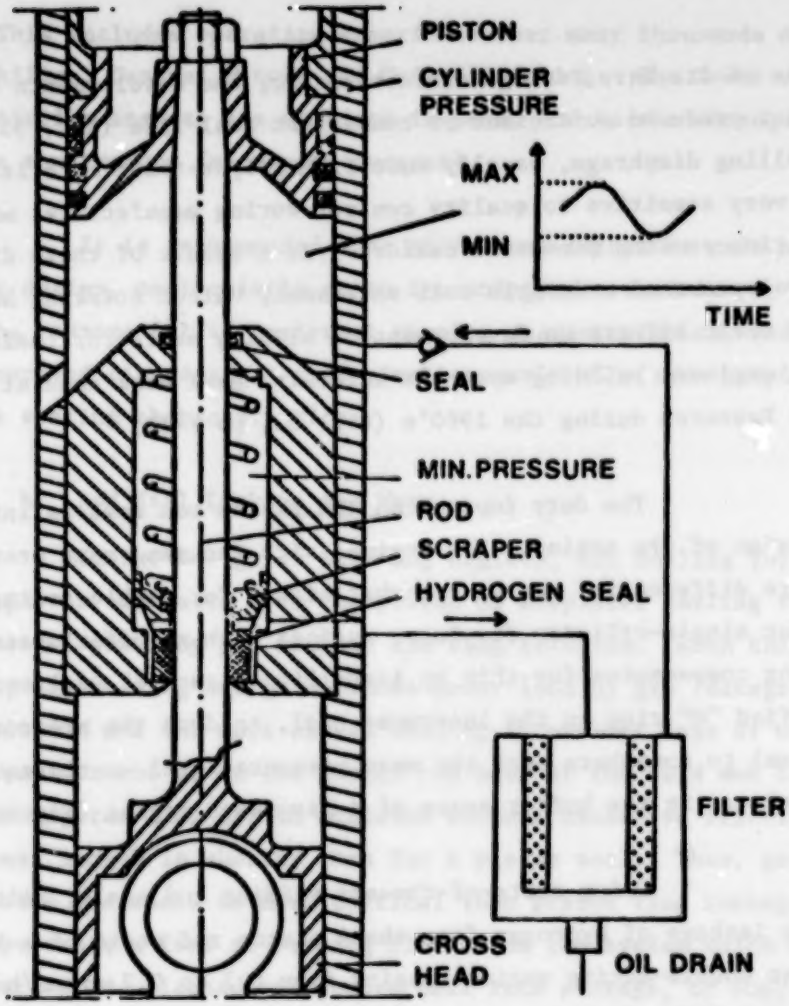

Figure 5-7 United Stirling Piston Rod Seal

"The surface finish of the piston rod or shaft was found to be important, with the optimum between 6 to 8 microinches. Initial values under 3 microinches resulted in high leakage -- an order of magnitude greater in some cases. Surface finish after running for hundreds of hours often decreased to about 4 microinches; but this appeared to have no effect on leak rate. An axial lap was preferred. Shaft out-of-roundness should not exceed 50 microinches and axial wave limit should not exceed 15 microinches in the seal operating region. Hydrostatic type grinding spindles are preferable; centerless grinding should be avoided".

At present, piston rod, sliding seal friction losses are excessive, amounting to as much as 400 watts per seal in a 100 bhp double-acting engine operating at a mean pressure of about 140 atmospheres. The seals are highly loaded both for restricting gas leakage, as well as to prevent ingress of lubricant. This necessitates cooling of the piston rod by an oil spray, which adds to the engine cooling load. Losses in the sliding seals for the GM small displacer engines were about 50 watts per seal; but the design employing a modified "O" ring was not suitable for mean pressure in excess of about 50 atmospheres and was not effective in preventing oil migration. Future sliding seal designs may incorporate dual function features. At relatively low mean pressure, one kind of seal with lower friction losses might function in combination with an oil scraping ring. As engine loading increased (higher mean pressure), another kind of seal and oil control scraper would gradually take over. As the engines become larger, the clearance between the piston rod and seal (as well as piston ring clearance) will not increase in proportion to the engine scale factor γ, but more nearly as $\sqrt{\gamma}$, or even less. In addition, the seal length is increased in proportion to γ, and internal working space volume increases γ^3. Consequently seal leakage as a percent of engine power rating will decrease and the time for a given loss of mean pressure should decrease as $\gamma^{1.5}$.

It is recommended that additional R&D be devoted to the effects of these variables on seal leakage and life; also more innovative seal designs need to be conceived and developed.

5.2 Characteristics of Basic Engine

The Stirling engine may be considered to consist of two major subsystems; the basic engine or thermodynamic cycle and the heat source system. This section deals with the basic engine which includes the components required to implement the basic cycle, e.g., cylinders and pistons, heater, regenerator and cooler, working gas and controls. The choice of mechanical arrangement for the basic engine was discussed above. This section will begin with the selection of the working gas, followed by a discussion of engine scaling rules which will permit the basic design parameters of large stationary engines to be determined by means of scaling up the size of existing automotive/truck engines. After presenting typical design specifications for a 2000 hp stationary engine, its performance characteristics will be estimated by applying the scaling rules. Then, the latter part of this section will describe the design features and problems encountered in the heat transfer components of the basic engine, i.e., heater head, regenerator, and cooler. Finally, the engine control methods are described and several miscellaneous characteristics of the basic engine are discussed.

5.2.1 Working Gases

The major considerations in the selection of the working gas for a Stirling engine are:

1. Thermodynamic, hydrodynamic and heat transfer characteristics.

2.4.2.3 Seal Design

The function of the rod seal in a Stirling engine is to contain the working fluid in the cylinder by preventing leakage into the crankcase, as well as to preclude the migration of crankcase contained fluids into the working space. The problem is complicated by the fluctuating high pressure of the working fluid, by slight lateral translation of the rod during the stroke, by the long life requirements under reciprocating conditions and by the sensitivity of the engine hot side to relatively small quantities of crankcase fluids.

Two basic approaches have been followed in the past; a positive seal consisting of a rolling diaphragm element and any of several forms of rubbing or sliding seals. The rolling element or rollsock seal has been pursued by N.V. Philips, with some test results exceeding 10,000 hours of life, but many failures at shorter times prevent the seal from being credited with a reliability which would allow it to be used in this state of the art program. Factors such as debris in the seal area can drastically reduce seal life, and extreme care and high skill levels are required for installation and servicing. Additional complications arise from the use of a pressurized oil backup system to reduce the tensile stress on the diaphragm. Thus this seal system is complex and vulnerable and is prone for sudden catastrophic failures unsuitable for stationary cogeneration systems.

Sliding seals fall into many categories, and combinations of types can be selected to provide specific sealing requirements. Gland or compression packings exert a mechanical force between the surfaces to be sealed and generally provide an adjustment which distinguishes them from the molded packings, including lip type and squeeze type seals. These seals function by providing one or more surfaces in contact with the reciprocating member, and in general can be fabricated from materials which permit them to function without supplementary lubricati

In addition to these seal types, a metallic scraper or pumping ring has been developed, and depending on its orientation, can function in either capacity. This device consists of a metal ring with the inner bore slightly tapered with the smaller end being nearly equal to the shaft diameter as shown in Figure 2.4.2-12. If oil is drawn into the bevel side, hydrodynamic forces cause the ring to be slightly expanded and oil is dragged through the gap. If oil exists on the side opposite the bevel, the scraping action of the sharp lip precludes the flow of oil into the ring. In practice, these seals can be used individually to serve either purpose, but are often used in opposed sets to maintain a reservoir of pressurized oil between them. The pumping ring is placed facing the oil volume of the engine and pumps oil into the cavity between it and the scraper ring, which acts as a termination of the intermediate reservoir. This approach has been used successfully as a method of eliminating the oil pump in rollsock seal systems, and pressures of over 1000 atmospheres have been reported.

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SLIDING SEAL SYSTEMS

Some information has been released on at least two systems which have been developed and tested in Stirling engines. The United Stirling seal is shown in Figure 2.4.2-13 and consists of an upper self-lubricated-molded seal whose purpose is to significantly mitigate the pressure excursions within the cycle, but not to serve as a positive seal. This "Kapseal" has been developed by N.V. Philips and exhaustively tested. Below this seal is a chamber which is pressurized with working fluid and contains a compression spring to provide axial force to the lower seal cartridges. These consist of a metal scraper seal to impede oil transport into the chamber, and a working fluid seal element below an intermediate pressure tap. This element passes some oil and some working fluid, which are drawn off using the pressure tap, filtered and the working fluid is returned to the engine space. As reported in the October, 1979 U.S. DOE Highway Vehicle Systems Contractors Coordination Meeting, efforts are underway to improve the oil sealing function of this seal concept. The problem reported is oil migration along the rod to the upper seal, and one method of eliminating this problem is being developed. It consists of a rubber diaphragm between the rod and seal housing and also prevents wear debris from settling onto the lower seals. Testing of the system must include the vulnerability of the diaphragm, increased assembly and maintenance time, and difficulty in determining when and if an internal failure has occurred (diaphragm leakage).

The second sliding seal system is the Ford/N.V. Philips seal, which contains four sealing elements. From the working fluid side, the first seal is a cap seal (may be same as Kapseal mentioned previously) to provide relatively constant pressure in the adjacent spring chamber. The next seal is a scraper seal of unorthodox configuration which is loaded by the axial spring. The primary sliding seal is next, and it uses the wedging action of the spring as well as the pressure drop it maintains to enhance rod preload. The fourth and last seal is a typical scraper ring which is adjacent to an oil filled cavity which provides rod cooling by introducing a flow of oil which bleeds out through a lower rod gap. An oil and working fluid outlet tap is provided in the area of the primary sliding seal. Pressure in that area is maintained just above the minimum cycle pressure by means of a working fluid check valve. The fluids recovered from the seal are processed through an oil separator and then the working fluid through a demister to condense any oil vapor or mist remaining. The oil collects in a float relieved reservoir which returns it to the crankase. Figure 2.4.2-14 shows this seal.

PROPOSED LSSE SEAL

For a stationary engine application, the design of the seal must consider not only seal reliability, but degraded mode operation in order to minimize forced outages and servicing of the seal and engine as well. Since the size is significantly larger than engines for which existing seals have been designed, some unique design problems (such as stroke length and significant rod lateral motion) exist. Also, due to the large power output per cylinder, the losses encountered in accepting a finite leakage through the seal system may become acceptable, permitting some variation in seal design.

The design shown in Figure 2.4.2-15 is proposed as a conceptual approach to meeting the LSSE requirements. The design consists of a Kapseal (or equivalent) at the working fluid end, again used to eliminate the pressure wave variations from the internals of the seal cartridges. A small buffer volume is next in line with a gas inlet and a labyrinth seal following. This labyrinth seal serves as a series back-up to the Kapseal for isolating pressure variations and may also serve as a substitute for a finite period of time (at higher power losses) until a failed Kapseal can be replaced. Also, the labyrinth assists in preventing oil migration into the working space by providing high gas velocities relative to the rod (especially in the upstroke direction when transport is most likely to occur) to produce a scrubbing action. The labyrinth is positioned relative to the rod by means of a floating ring to accommodate rod eccentricities. Below the seal is a chamber of adequate length to assure that the portion of the rod that contacts oil in the lower seals does not enter the labyrinth area.

A fluid removal line is used to extract any working fluid (either leaked past the Kapseal or added above the labyrinth to promote flow) and oil that may exist in the volume. This mixture of helium and oil vapor or mist is processed through an oil mist/vapor separator shown in Figure 2.4.2-16 and Section 2.7.3 and returned to the working space and oil reservoir respectively.

The next and final seal is a scraper seal, also in a laterally floating housing. This scraper removes most oil from the rod before it is carried into the seal cavity. Immediately below the scraper is a cavity into which oil is pumped from a remote source to provide cooling to the rod and also to reduce the pressure drop across the seal and consequently reduce working fluid loss to the crankcase. The fit of the outlet with respect to the rod provides the desired internal pressure and adequate oil flow. The seals are housed in a cartridge to facilitate replacement or servicing, and a lower flange provides access to the internal chambers via ports. The pressure induced axial force is carried through the flange and reacted in the external flange bolts.

Approximate power consumption predictions for this seal system are 0.32 horsepower for cooling oil pumping and .5 horsepower for labyrinth seal flow pumping, using a seal chamber pressure of 88 atmospheres. Main objective of this seal design approach is to avoid forced outages by providing a capability to tolerate seal leakage, without shutdown. Gas flow indicators in individual cylinder return lines can monitor the state of the seal and thus provide information on necessary scheduled maintenance.

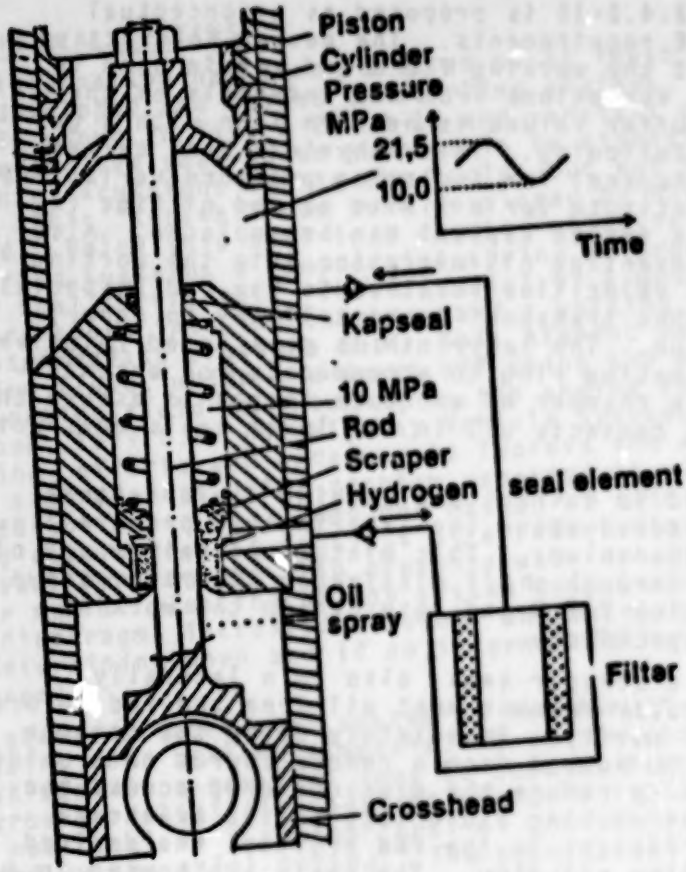


Figure 2.4.2-13--United Stirling Sliding Seal

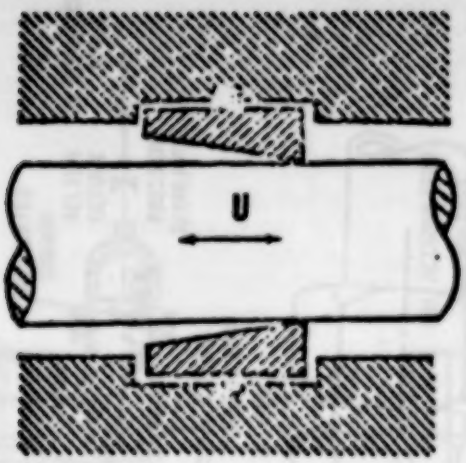
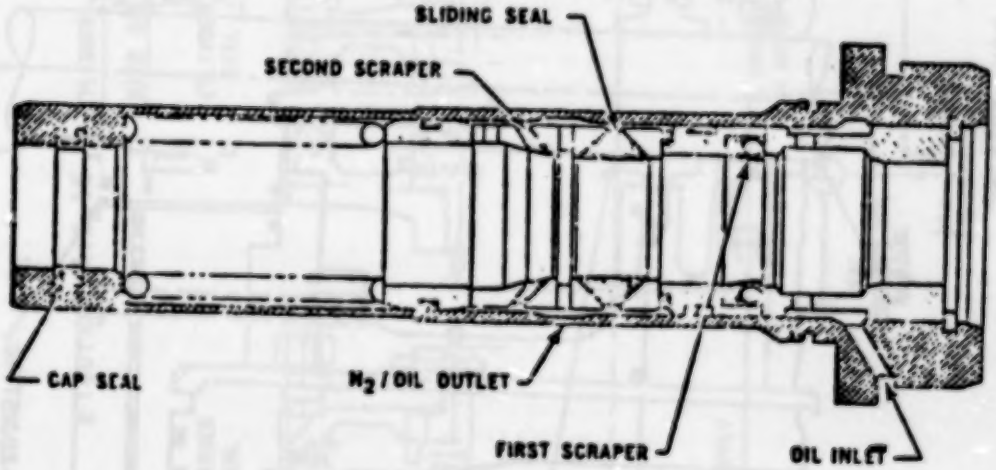


Figure 2.4.2-12 Scraper Seal (or Pumping Ring)



SLIDING SEAL CARTRIDGE CROSS SECTION

Figure 2.4.2-14 Ford/N.V. Philips Sliding Seal

FIGURE 2-15

PISTON ROD SEAL CONCEPT

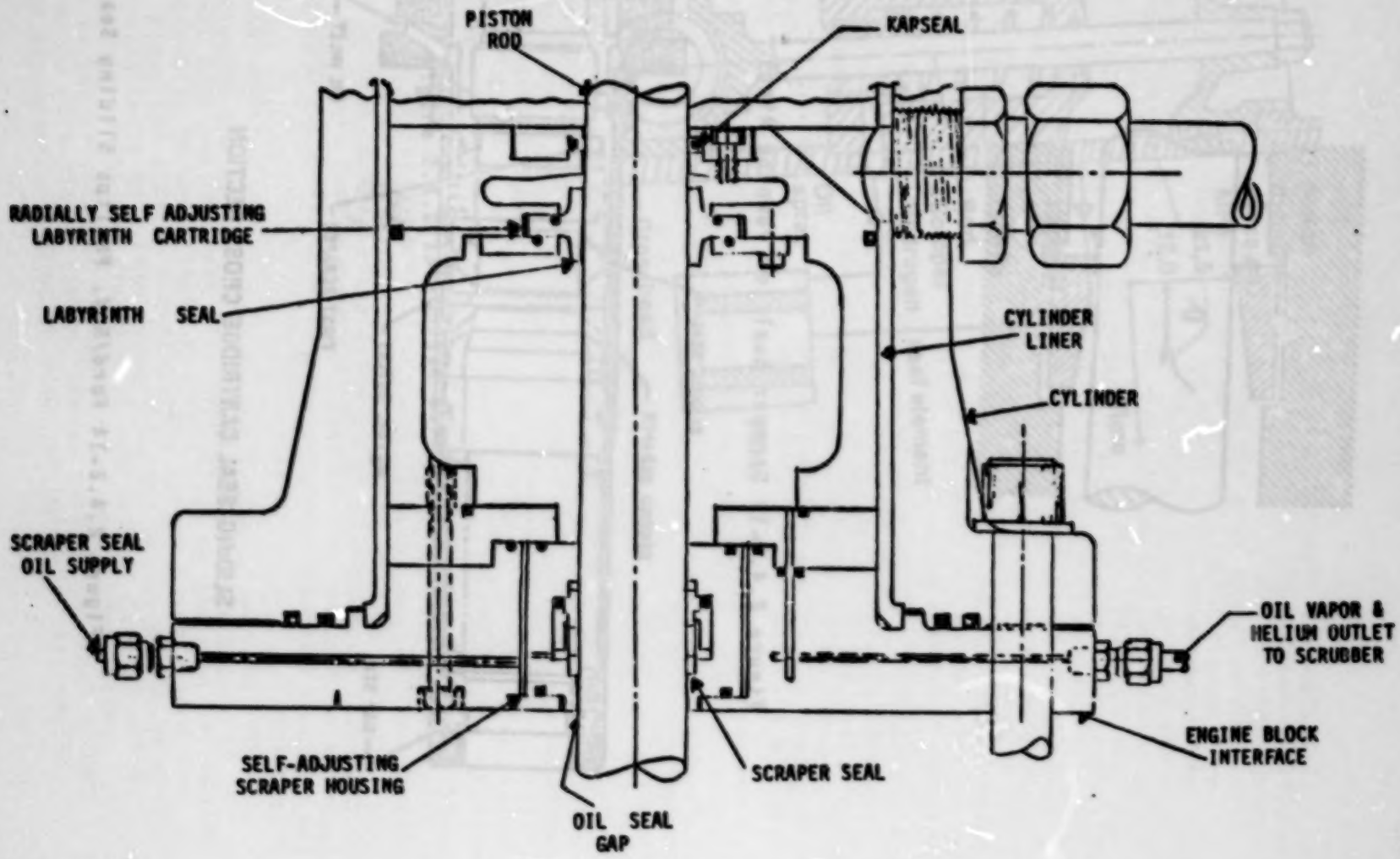
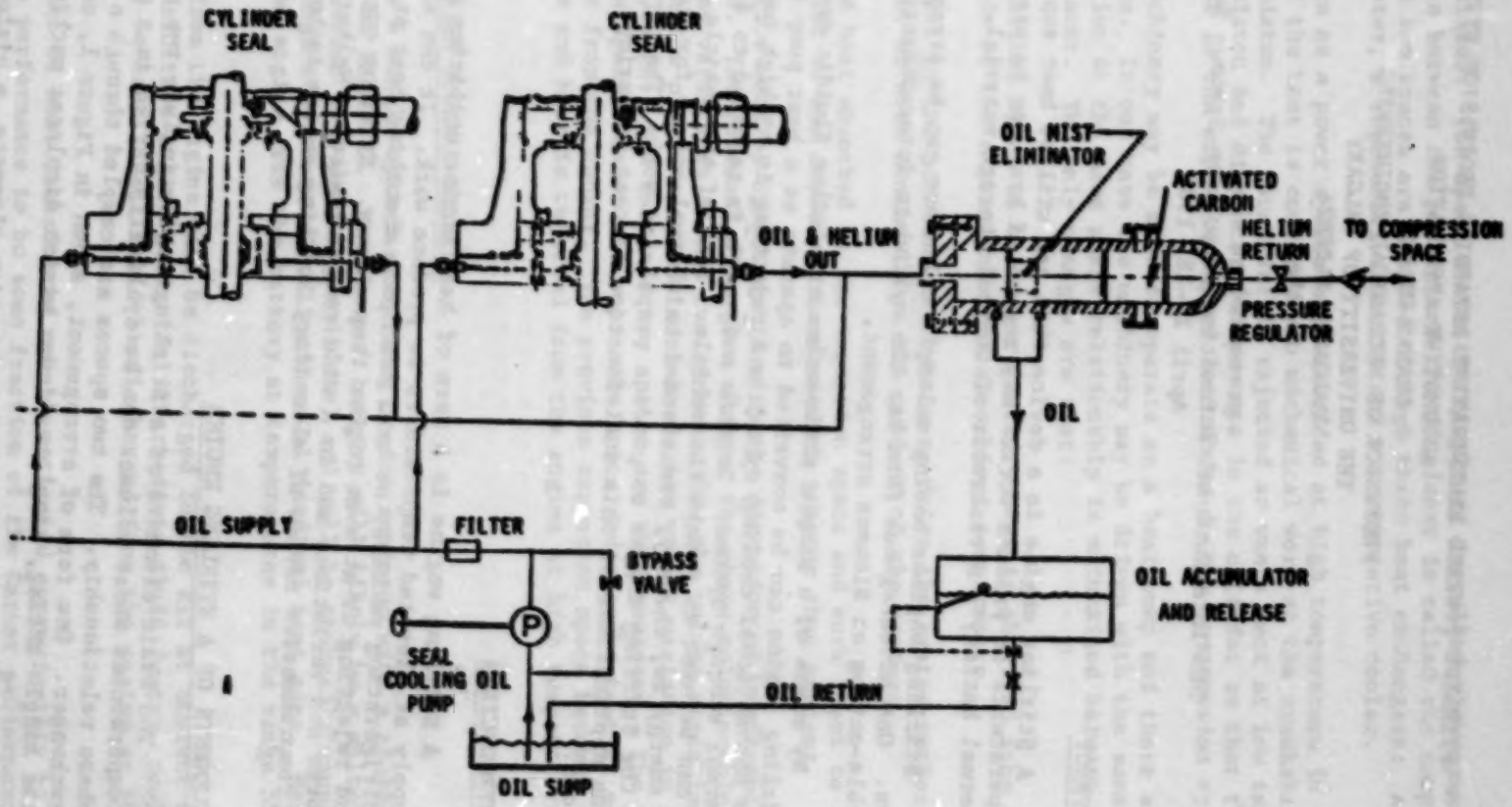


Figure 2.4.2-16 CYLINDER SEAL SYSTEN WITH OIL ELIMINATOR



STIRLING POWERED REGENERATIVE RETARDING PROPULSION SYSTEM FOR AUTOMOTIVE APPLICATION

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presented at the
5th International Automotive Propulsion Systems Symposium
April 14-18, 1980

ABSTRACT

A Stirling engine is a device that can utilize heat supplied at elevated temperatures to produce work. Any source of heat can be used including a 'thermal battery', a reservoir of thermal storage material.

Stirling engines having multiple cylinders can be arranged in a number of ways. One advantageous form has the cylinders interconnected in the so-called double-acting or Siemens arrangement.

By means of a simple slide-valve or similar fluidic device a Siemens Stirling engine can be converted to operate as a heat pump drawing heat from the conventional cooling system and rejecting it at high temperature to the thermal storage system. In this mode work is required to drive the engine so it can be used to retard the vehicle. Part of the vehicle potential or kinetic energy is, thereby, transformed to thermal energy for subsequent utilization in the Stirling engine to produce propulsion work. The paper will consider some applications of this dual-function concept to various automotive applications.

INTRODUCTION

A Stirling engine is a form of heat engine utilizing an external heat supply at elevated temperatures to produce work. It can also operate as a refrigerating machine, or as a heat pump absorbing heat at ambient temperature and rejecting it at some higher temperature. Stirling engines exist in many forms and variations and in a wide range of sizes. Walker (1980)* has provided a comprehensive review of elementary Stirling engine technology.

ELEMENTS OF A STIRLING ENGINE

The basic elements of a Stirling engine include two spaces at different temperatures whose volumes can be cyclically varied in a constant but out of phase relationship. The two spaces are coupled through a regenerative heat exchanger. One form of arrangement, shown in Figure 1, can be described as the single acting, single cylinder piston displacer machine.

It consists of a two reciprocating elements, a piston and a cylinder

* Complete reference given at the end of the text.

working in a cylinder. The space above the displacer is called the expansion space. The space between the piston and the displacer is called the compression space. The two spaces are coupled through three heat exchangers: a recuperative heater, a regenerative matrix and a recuperative cooler.

In operation as a power system heat is added at high temperature in the heater. Some of the heat is converted to mechanical work at the crankshaft coupled to the piston. The remainder is rejected as waste heat at low temperature. The piston and displacer reciprocate in the cylinder so that the volume variations in the expansion space lead those in the compression space.

The same machinery may be made to operate as a heat pump and there are two possibilities. In one case the machinery may be driven with the same direction of rotation so the same phase relationship is maintained between the piston and displacer. The only changes are that:

a) the expansion space is no longer heated. Heat is absorbed from the surrounds of the expansion space at ambient temperature.

b) work input to the piston is necessary to drive the system rather than an output of work available as the product of the process.

The combined heat absorbed in the expansion space and work input to the system are rejected to the cooler at temperatures greater than ambient.

The alternative mode of operation is with the direction of crank-shaft rotation reversed. The reversal of motion changes the phase relationship between the volume variations in the two spaces. Now the volume variation in the space between the piston and displacer leads the volume variation in the space above the piston. The roles of two spaces are thus reversed. The previous compression space becomes the expansion space and absorbs heat at ambient temperature from the cooler. The previous expansion space becomes the compression space and heat is rejected from the engine at high temperature in the heater.

STIRLING ENGINE WITH THERMAL BATTERY

Consider now a single cylinder Stirling engine equipped with a thermal battery as shown in Figure 2. The thermal battery will likely be a molten salt (lithium fluoride is a popular candidate) and will be thermally coupled to the Stirling engine by means of a liquid metal (sodium) heat pipe. Heat is supplied to the engine from the battery at temperatures in the range 550 to 850°C.

Waste heat from the engine will be discharged to the air at ambient temperatures. To facilitate engine design an intermediate fluid (water) cooling loop would likely be used. Heat would be discharged from the engine at 60 to 80°C.

The quantities of heat transferred can be estimated approximately by assuming the engine performance to be some fraction of the Carnot performance. The assumption of a performance equivalent to half the Carnot would not be

unreasonable.

$$\text{Therefore: Thermal efficiency } \eta = \frac{\text{Work output } (W_o)}{\text{Heat input } (Q_I)}$$

$$= \frac{\text{Heat input } (Q_I) - \text{Heat rejected } (Q_o)}{\text{Heat input } (Q_I)} = 1 - \frac{\text{Heat rejected}}{\text{Heat input}}$$

$$\text{but } \eta = 0.5\ \eta_{Carnot} = 0.5 \left(\frac{T_{max} - T_{min}}{T_{max}} \right).$$

where η = thermal efficiency
W_o = work output of engine
Q_I = heat supplied to engine from thermal battery
Q_o = heat rejected from the engine to cooling water
η_{Carnot} = theoretical Carnot efficiency
T_{max} = maximum cycle temperature (at which heat is supplied to the engine)
T_{min} = minimum cycle temperature (at which heat is rejected from the engine)

Now let T_{max} = 850°C (1123K) and
T_{min} = 80°C (353°K).

Then $\eta = 0.5 \left(\frac{1123 - 353}{1123} \right) = 34$ percent.

Per kW of engine output 2.9 kW (say 3 kW) of heat would be drawn from the thermal battery and 1.9 (say 2) kW would be rejected to the cooling system.

STIRLING ENGINE WITH THERMAL BATTERY OPERATING AS A HEAT PUMP

Consider now the same Stirling engine operating as a heat pump complete with thermal battery, sodium heat pipe and water cooling fluid loop.

To change the mode of operation from a power system to a heat pump we simply stop the engine and drive it in the reverse direction, thereby switching the location of the expansion and compression spaces.

The engine will draw heat from the cooling water and reject heat at high temperature to the thermal battery. Of course work input will be required to drive the engine. In vehicular applications this work input could come from the kinetic energy of the vehicle or potential energy when running downhill.

For the sake of simplicity let us keep the same temperature distribution (850 and 80°C) and also assume a Coefficient of Performance of the Stirling heat pump to be half that of a Carnot heat pump.

Then: Coefficient of Performance $COP = \dfrac{\text{Heat Delivered at high temperature } (Q_D)}{\text{Work Input } (W_I)}$

$$= \frac{\text{Heat Delivered } (Q_D)}{\text{Heat Delivered } (Q_D) - \text{Heat Lifted } (Q_L)}$$

But $COP = 0.5\ COP_{Carnot} \left(\dfrac{T_{DEL} - T_{LIF}}{T_{DEL}} \right)$

where COP = Coefficient of Performance
Q_D = Heat delivered to the thermal battery at high temperature
Q_L = Heat lifted from the cooling water at low temperature
W_I = Work input to the engine
COP_{Carnot} = Theoretical Carnot Coefficient of Performance
T_{DEL} = Temperature at which heat is transferred from the engine to the thermal battery
T_{LIF} = Temperature at which heat is transferred from the cooling water to the engine.

Now if T_{DEL} = 850°C (1123K) and
T_{LIF} = 80°C (353K)
then COP = $0.5 \left(\dfrac{.1123}{1123 - 353} \right) = 0.73.$

For every kW of work delivered to the engine 0.73 (say 0.75) kW of high temperature energy would be delivered to the thermal battery for subsequent use in the engine power mode.

In the case of a locomotive or freight truck running on the level every kW of engine output required to accelerate to operating speed would require the drain of 3 kW from the thermal battery. When stopping the power consumed to accelerate the vehicle could be reconverted to high temperature heat using the engine as a heat pump. Assuming no losses due to friction and windage etc. the 1 kW of work input to the engine would result in 0.75 kW to the thermal battery. This would provide 0.25 kW of work for the next start, a saving in fuel expenditure of 25 per cent.

When descending grades of course the potential energy (normally absorbed by frictional dissipation in the brakes) could be converted to high grade heat in the thermal battery for use in driving up the next hill.

ACTUAL PERFORMANCE

The assumed values of half-Carnot performance are conservative. This level of performance in Stirling engines or cryocoolers is routinely reported in the literature. Given the stimulus to strive for maximum performance it is possible that 0.75 of the Carnot performance could be attained.

In that situation $\eta = 0.75 \left(\dfrac{1123 - 353}{1123} \right) = 0.51$

and $COP = 0.75 \left(\dfrac{1123}{1123 - 353} \right) = 1.09.$

Every 1 kWh of work input to the heat pump results in 1 kWh going to the thermal battery and half that could be subsequently generated as work in the engine in the power mode. The fuel saving with regenerative braking sample level accelerating and decelerating is 50 per cent.

THERMAL CAPACITIES

Meijer (1970) presented the data reproduced in Figure 3 showing the energy content and potential mechanical power available from a thermal battery in combination with a Stirling engine. The maximum energy density at 850°C is 0.47 kWh/kg of lithium fluoride. The mechanical energy potentially available is 0.2 kWh per kg of lithium fluoride.

For comparison diesel fuel with a calorific value of 20000 BTU per lb. has a thermal equivalent of 12.8 kWh/kg and, in an engine of 40 percent thermal efficiency, a potential mechanical energy of 5.2 kWh. It must be recalled that diesel fuel consumes 15/20 times its own mass of air and is not rechargeable as the thermal battery is but is simply consumed.

AUTOMOTIVE STIRLING ENGINES

The power requirements for automotive applications extends from 20 kW (small cars) to 1000 kW (locomotives and large mining trucks). Stirling engines could embrace this power range but none would have just a single cylinder; four, eight or sixteen cylinders would be more likely. They could consist simply of an ensemble of single cylinder engines mounted in an inline or vee arrangement on a common crankshaft. Early Stirling engines for vehicular use were of this type.

These have now been supplanted by an arrangement in which the multiple cylinders are interconnected as shown in Figure 4. Devised about 1860 by Sir William Siemens (Walker (1980)), it is used for all the present automotive engine developments. Figure 5 shows the customary four cylinder arrangement and Figure 6 is the well-known United Stirling engine said to be about ready for production.

It will be noted from Figure 4 that in the Siemens arrangement there is only one reciprocating element per cylinder instead of the customary piston and displacer. The expansion space of one cylinder (above the piston) is coupled through the heat exchangers to the compression space (below the piston) of the adjacent cylinder.

The system can be operated as a power system or as a heat pump just the same as the single cylinder machine. One way is to reverse the direction of rotation of the engine. This causes the roles of the spaces above and below the piston to reverse and so reverse the direction of the heat flow.

Complicated heavy-duty transmission arrangements would be necessary to reverse the direction of rotation of the engine. It is both undesirable and unnecessary for exactly the same effect can be gained by a simple fluid control valve, which allows the engine to operate *in the same direction of rotation* as a power system and as a heat pump.

The system is illustrated in Figure 7. Figure 7 (a) is a reproduction of Figure 4 showing the engine operating as a power system. The upper spaces of the engine cylinders are coupled to the lower spaces of adjacent cylinders to their *right*.

To convert to operation as a heat pump and retain the same forward direction of rotation it is necessary only to couple the upper spaces of engine cylinders to the lower spaces of adjacent cylinders to their *left*. This simple fluid switch reverses the roles of the upper and lower spaces causing a reversal in the direction of heat flow.

The same effect could of course be achieved by flow switching of the upper spaces but this would require a high temperature device and so switching in the low temperature region of the engine is preferred.

Figure 8 shows a conceptual view of the fluid connections required to effect the conversion from power system to heat pump. This could readily be incorporated in the same power control system already used on some engines to interconnect all cylinders thereby 'spoiling' the pressure characteristic and effecting an instantaneous decrease in engine power. Three conceptual forms of switching valve, a shuttle valve, rotary valve or disc valve are shown in Figure 9.

APPLICATIONS

Present research on Stirling engines for automotive applications appears devoted entirely to engines using diesel fuel. This is unfortunate. The compression ignition engine is the best use for diesel fuel as the spark ignition engine is for gasoline.

The real automotive applications for Stirling engines lie ahead in the era, now close at hand, when diesel fuel and gasoline are no longer available at low cost. In that future time the sources of energy for automotive propulsion will include electricity, natural gas and coal and the various agricultural, municipal, industrial, commercial and residential wastes.

The thermal battery in combination with a Stirling engine will then come into its own. The thermal battery will be charged overnight by electric resistance heating or by combustion of natural gas, coal or other solid fuel combustion. The system could be applied to commuter cars, municipal buses, taxi fleets, delivery and commercial vehicles, farm tractors and underground mine vehicles.

The regenerative retarding proposals contained here would be particularly attractive for municipal buses and delivery vehicles with their frequent stops and starts and would likely be found worthwhile on other vehicles for urban use because of the intermittent nature of city traffic.

Automotive applications where overnight recharging would not be feasible include railway locomotives, long distance trucks, and heavy surface mine trucks. In these applications the ability to burn coal as the vehicle fuel is attractive and could be accomplished with a fluidized bed reactor. There are compelling

technical reasons to interpose a liquid metal (sodium) heat pipe between the coal combustor and the Stirling propulsion engine. Given this heat pipe/Stirling combination little expense is required to add the thermal storage system to capitalize on the opportunity for regenerative braking during stops or in descending grades.

For surface mine trucks engaged in loaded downhauls very significant savings in fuel are possible. The potential energy of the truck and payload can be converted to thermal energy for storage and is recovered as power during ascent of the empty truck to reload. Depending on the conditions assumed savings of up to 60 per cent of the fuel necessary to drive the truck up the hill can be gained. The savings can be phenomenal with a fleet of 50 trucks having 1000 horsepower engines working 22 hours a day in remote areas with high fuel costs. In coal mines the opportunity to run the truck on coal instead of oil is compelling.

For mine locomotives engaged in loaded uphaul duty involving part operation on the surface and part operation underground the thermal storage/Stirling engine system is attractive to eliminate exhaust pollutants underground. The thermal storage system can be recharged by coal or oil combustion during surface operation. For underground operation the locomotive is energized from the thermal store. Conversion of potential energy of the unloaded train by regenerative braking supplements the thermal storage but of course is nowhere near as attractive as the loaded downhaul applications.

FURTHER INFORMATION

We have given here a simplistic exposition of a complicated topic with no technical detail and leaving many questions unanswered. For those who wish to pursue the subject further the book by Walker (1980) contains much useful information and many references. One of the earliest and still the most significant reference to the Stirling engine/thermal storage combination was given by Meijer (1970).

Research reports and current developments about Stirling engines may be found in the proceedings of the Intersociety Energy Conversion Engineering Conference (Proc. IECEC) held annually (in August) in the United States.

REFERENCES

WALKER, G. (1980), Stirling Engines, Oxford University Press.

MEIJER, R. (1970), Prospects of the Stirling Engine for Vehicular Propulsion, Philips Tech. Rev. 31, pp. 168-185.

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Figure 8: Cylinder connections for the Siemens engine as a power system and heat pump.

Figure 9: Conceptual Forms of Flow-switching valve, shuttle, rotary and disc.

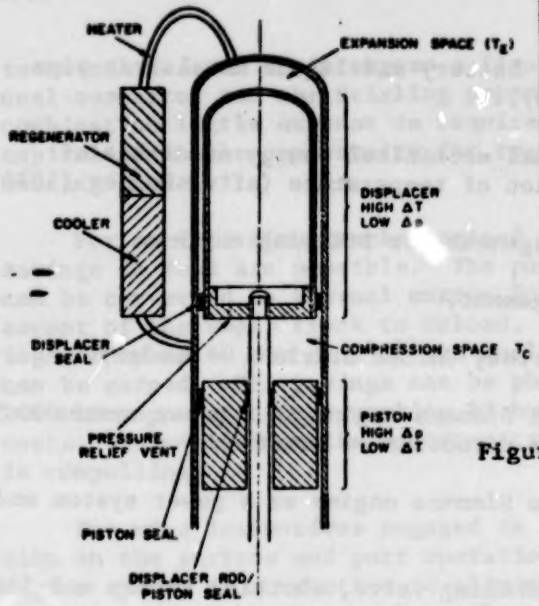

Figure 1: Single Acting, Single Cylinder, Piston and Displacer Stirling Engine.

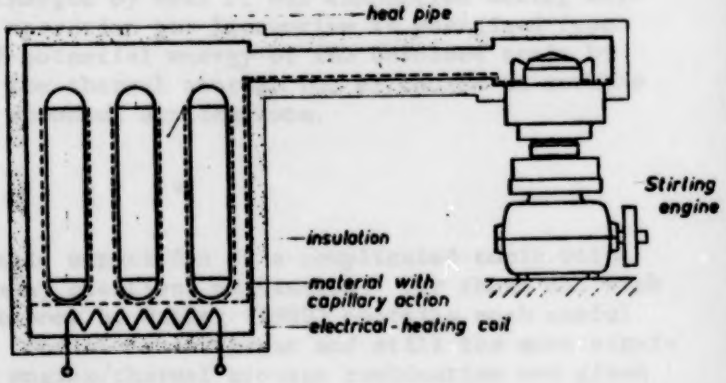

Figure 2: Stirling Engine with Thermal Battery and Liquid Metal Heat Pipe Coupling (after Meijer (1970)).

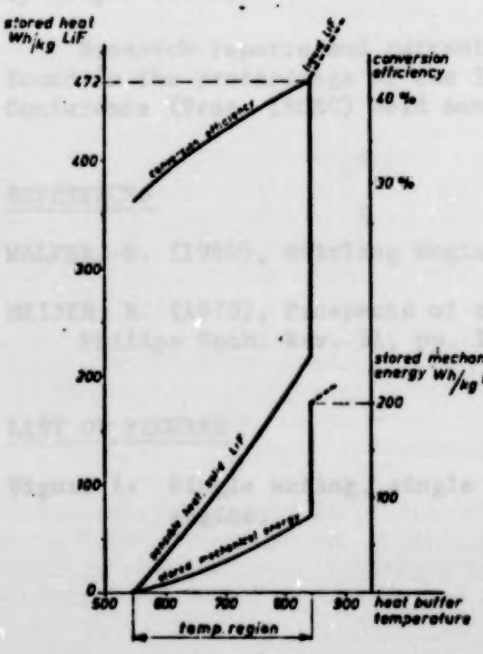

Figure 3: Thermal Capacity and Potential Mechanical Energy Available of Lithium Fluoride as a Function of Temperature (after Meijer (1970)).

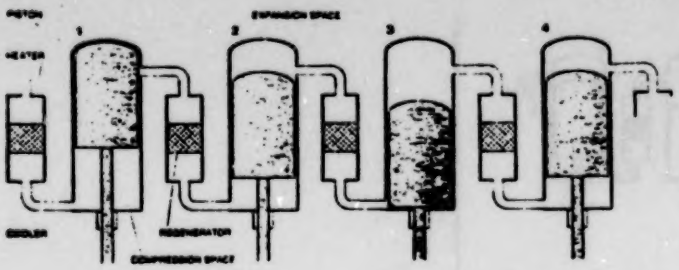

Figure 4: Siemens Multi-Cylinder Arrangement for Stirling Engines.

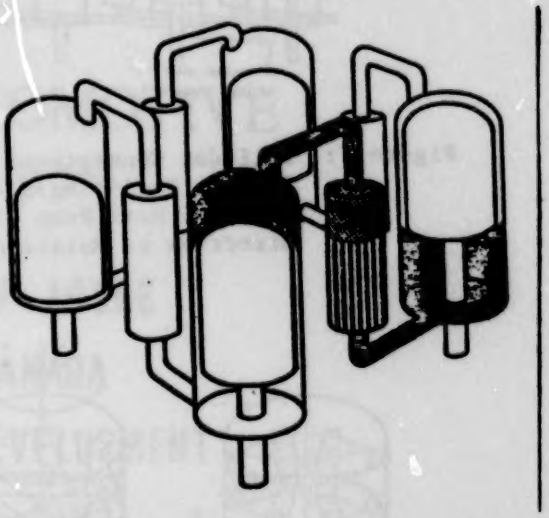

Figure 5: Four Cylinder Siemens Arrangement.

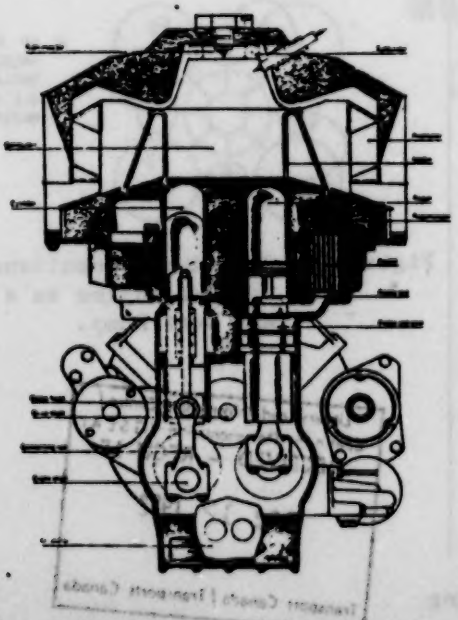

Figure 6: United Stirling Engine (Courtesy United Stirling - Sweden).

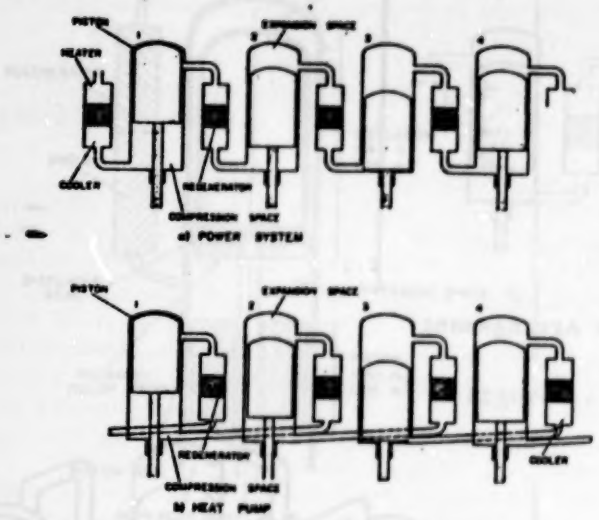

Figure 7: Cylinder Connections for the Siemens Engine as a Power System and as a Heat Pump with the Same Direction of Rotation.

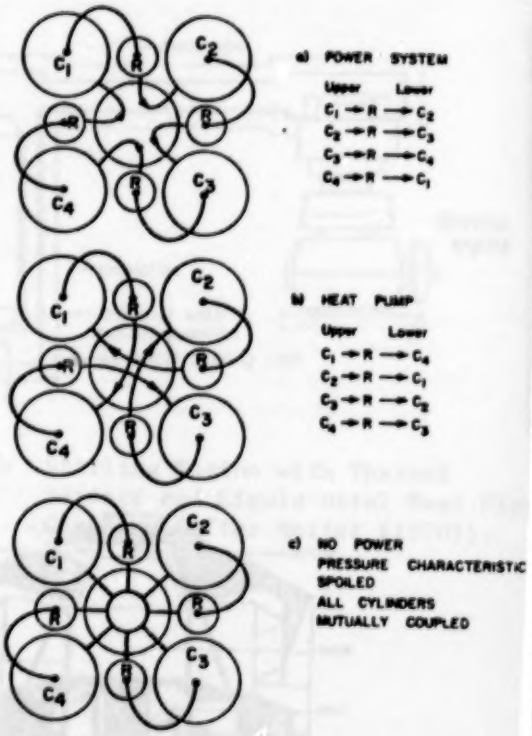
a) POWER SYSTEM
Upper Lower
C1 → R → C2
C2 → R → C3
C3 → R → C4
C4 → R → C1
b) HEAT PUMP
Upper Lower
C1 → R → C4
C2 → R → C1
C3 → R → C2
C4 → R → C3
c) NO POWER
PRESSURE CHARACTERISTIC SPOILED
ALL CYLINDERS MUTUALLY COUPLED

Figure 8: Cylinder Connections for the Siemens Engine as a Power System and Heat Pump.

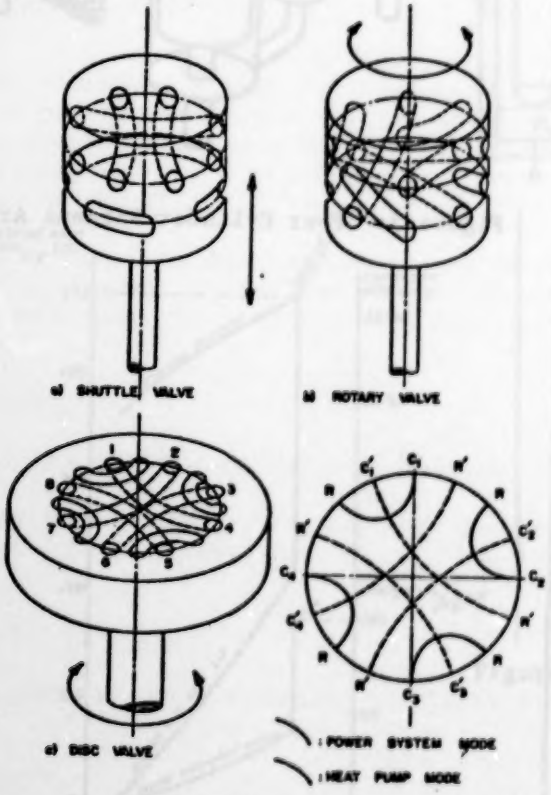

Figure 9: Conceptual Forms of Flow-Switching Valve, Shuttle, Rotary and Disc.

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THE STIRLING ENGINE FOR RAILWAY LOCOMOTIVE APPLICATION

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THE STIRLING ENGINE FOR RAILWAY LOCOMOTIVE APPLICATION

by

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Transport Canada
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Montreal
Quebec
Contract No. 4270

February 1981

Report No. T.P. 3242

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REPORT DOCUMENTATION FORM

1. Transport Canada Report No.	2. TDC Project No.	3. Recipient's Catalogue No.
T.P. 3242	4270	

4. Title and Subtitle	5. Report Date
The Stirling Engine for Railway Locomotive Application	
	6. Performing Organization Report No. None Given
7. Author(s)	8. Transport Canada File No.
G. Walker	D500-642-1
9. Performing Organization Name and Address	10. DSS File No.
Whitegaites Engineering Ltd. 4023 Vardell Road N.W. Calgary, Alberta T3A 0C3	11. DSS or Transport Canada Contract No. 4270
12. Sponsoring Agency Name and Address	13. Type of Report and Period Covered Summary Dec. 80 - Feb. 81
Transportation Development Centre 1000 Sherbrooke St. W. P.O. Box 549 Montreal, Que. H3A 2R3	14. Sponsoring Agency Code
15. Supplementary Notes	16. TDC Project Officer R.H.F. Dillon

17. Abstract

Future shortages and high costs of fuel oil are expected to result in the use of coal for railroad locomotives. A brief study has been made of the Stirling engine for this application. A Stirling engine is a heat engine receiving heat from an external source, converting a fraction of it to useful work and rejecting the remainder as waste heat. It was invented in the 19th century and was widely used until displaced by the internal combustion engine and electric motor at the turn of the century.

Redevelopment of the Stirling engine began about 40 years ago. It is widely used for cryogenic cooling and advanced prototypes up to 150 kW have been developed for automotive use.

Conceptual engineering studies of large (1000 horsepower) Stirling engines have been made in the United States. A comparison of these with contemporary locomotive diesel engines shows that Stirling engines of equivalent power and efficiency could fit the same installation envelope as direct replacement power units in existing locomotives.

Fluidized bed coal combustors would be located beneath the locomotive frame where the oil fuel tanks are normally located.

Novel proposals are made for Ringbom-Stirling engines using air as the working fluid and water as the lubricant with power boost at maximum power output by high pressure steam injection. Such engines could be made by converting existing locomotive diesel engines using the same parts and components. Enormous savings would result from the gradual conversion of existing locomotive fleets to coal-burning operation rather than outright replacement as would be required for steam locomotives.

18. Key Words	19. Distribution Statement
Stirling, Engine, Railway, Locomotive, Air Engine, Coal, Fluidized Bed, Heat Pipe	A limited number of reports are available.

20. Security Classification (of this report)	21. Security Classification (of this page)	22. No. of Pages	23. Price
Unclassified	Unclassified		

TDC/CDT 79-005

REPORT DOCUMENTATION FORM

1. Transport Canada Report No.	2. TDC Project No.	3. Recipient's Catalogue No.
T.P. 3242	4270	

4. Title and Subtitle	5. Report Date
The Stirling Engine for Railway Locomotive application	6. Performing Organization Report No.

7. Author(s)	8. Transport Canada File No.
G. Walker,	D500-642-1

9. Performing Organization Name and Address	10. DSS File No.
Whitegaites Engineering Ltd. 4023 Vardell Road N.W. Calgary, Alberta T3A 0C3	11. DSS or Transport Canada Contract No. 4270

12. Sponsoring Agency Name and Address	13. Type of Report and Period Covered
Centre de développement des Transports 1000 ouest, rue Sherbrooke - 25e étage Montréal, Québec H3A 2R3	Summary, décembre 1980-février 1981 14. Sponsoring Agency Code

15. Supplementary Notes	16. TDC Project Officer
	R.H.F. Dillon

17. Abstract Les pénuries prévisibles et les coûts élevés du carburant provoqueront peut-être l'emploi du charbon dans les locomotives de chemin de fer. A cette fin, une étude sur le moteur STIRLING a été entreprise. Il s'agit d'un moteur thermique recevant sa chaleur d'une source extérieure, convertissant une partie de la chaleur en force thermique et rejetant le reste en perte de chaleur. Ce moteur a été inventé au 19e siècle et a été en usage jusqu'à la fin du siècle lorsqu'il a été remplacé par les moteurs à combustion interne et électrique. Ce moteur a de nouveau été utilisé depuis une quarantaine d'années. Il est largement utilisé pour la réfrigération et des prototypes élaborés jusqu'à 150 KW ont été créés pour les véhicules à moteur. Aux Etats-Unis, des études conceptuelles d'ingénirie du gros moteurs Stirling (1000 HP) ont été entreprises. Une comparaison avec les moteurs diesel contemporains indique que ces moteurs Stirling, à puissance et à capacité égales, pourraient s'adapter facilement pour remplacer les moteurs thermiques dans les locomotives présentement en usage. Les éléments de combustion du charbon sur un lit fluidisé seraient placés sour le chassis des engins de locomotive diesel où les réservoirs de mazout sont habituellement localisés. La nouveauté de l'idée consiste à utiliser un moteur Ringbom-Stirling qui utiliserait l'air comme fluide de travail et l'eau comme lubrifiant. La puissance de sortie maximale serait obtenue par suralimentation en injectant de la vapeur à pression élevée. L'avantage est que les locomotives diesel d'utilisation courante pourraient être converties aux moteurs à cycle de Stirling, tout en gardant le même emballage. Ainsi, il serait possible de réaliser de très fortes économies en dotant graduellement le parc actuel de tels moteurs, plutot que de remplacer carrément ce parc par des locomotives à vapeur.

18. Key Words	19. Distribution Statement
Cycle de Stirling. Moteur thermique. Chemin de fer, Locomotive. Moteur à vapeur Charbon. Lit fluidisé.	Le CDT dispose d'un nombre limité d'exemplaire.

20. Security Classification (of this report)	21. Security Classification (of this page)	22. No. of Pages	23. Price
Non classifée	Non classifée		

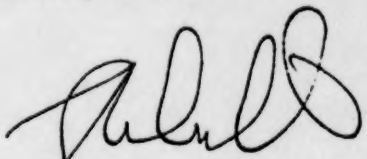
TDC CDT 79-005

ACKNOWLEDGEMENTS

Helpful information and advice about the Series 251 diesel engine was received from Bombardier Inc./MLW, Montreal, Quebec.

FOREWORD

This limited study of the potential for application of coal-burning Stirling engines to railroad locomotives in the context of Canadian transportation was carried out under contract with Transport Canada, Transportation Development Centre, Montreal, Contract No. 4270. The project officer for Transport Canada was Mr. Richard Dillon. Tne study was performed during December 1980/February 1981.



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SUMMARY

Future shortages and high costs of fuel oil are expected to result in the use of coal for railroad locomotives. A brief study has been made of the Stirling engine for this application.

A Stirling engine is a heat engine receiving heat from an external source, converting a fraction of it to useful work and rejecting the remainder as waste heat. It was invented in the 19th Century and was widely used until displaced by the internal combustion engine and electric motor at the turn of the century.

Redevelopment of the Stirling engine began at Philips Laboratories, Eindhoven, some 40 years ago. Now the Stirling engine is widely used for cryogenic cooling and advanced prototypes up to 150 kW have been developed for automotive use by the Swedish licensees, United Stirling.

Detailed engineering studies of large (1000 horsepower) Stirling engines have been made in the United States. A comparison of these proposed engines, modified for locomotive use with contemporary locomotive diesel engines shows that Stirling engines of equivalent power and efficiency could fit in the same installation envelope as direct replacement power units in existing locomotives.

Fluidized bed coal combustors would be located under the locomotive frame between the trucks where the oil fuel tanks are normally located. Sufficient space would remain for a small buffer coal store fed by screw auger from a coal tender towed by the locomotive.

Novel proposals are made for Ringbom-Stirling engines using air as the working fluid and water as the lubricant with power boost at maximum power output by high pressure steam injection. Such engines could be made by converting existing locomotive diesel engines using the same parts and components. Enormous savings would result from the gradual conversion of existing locomotive fleets to coal-burning operation rather than outright replacement as would be required for steam locomotives. Recommendations are made for further engineering designs of the Ringbom-Stirling engine including the development of a 20 horsepower proof-of-principle demonstration engine converted from an industrial two-cylinder diesel engine. Estimated cost and time of the programme is $250,000 per year for 3 years.

Part II of the report contains a brief appraisal of gas-fired Stirling engines for locomotive propulsion including some aspects of liquid energy gas handling, storage and the hazards attendant therein.

RÉSUMÉ

Les pénuries prévisibles de carburant et les coûts de plus en plus prohibitifs provoqueront peut-être l'emploi du charbon pour la propulsion des locomotives de chemins de fer.

Dans cette éventualité une étude succinte a été effectuée du moteur à combustion externe, ou moteur à cycle de Stirling. Il s'agit d'un moteur thermique recevant sa chaleur d'une source extérieure, convertissant une partie de la chaleur en force thermique et rejetant le reste en perte de chaleur. Ce moteur dont l'invention remonte au 19e siècle a été très répandu jusqu'à l'avènement du moteur à combustion interne et du moteur électrique, au début du 20e siècle.

Le moteur Stirling a été développé, il y a environ quarante ans par les laboratoires Philips, à Eindhoven, dans les Pays-Bas. De nos jours, il est largement utilisé pour la réfrigération cryogénique et des prototypes élaborés d'une puissance de 150 KW ont été créés pour les véhicules à moteur par United Stirling, concessionnaire suédois.

Aux États-Unis, des études détaillées ont été effectuées, portant sur des moteurs Stirling capables de développer (1000HP). Une comparaison de ces moteurs avec les moteurs diesel contemporains indique que les moteurs Stirling, à puissance et à capacité égales, pourraient s'adapter facilement pour remplacer les moteurs thermiques dans les locomotives présentement en usage, tout en occupant le même volume d'espace.

Les éléments de combustion du charbon sur un lit fluidisé seraient placés sous le châssis de la locomotive, là où les réservoirs de mazout sont habituellement installés. Il resterait assez de place pour un petit magasin à charbon, servant de tampon et alimenté par vis sans fin à partir du tender.

La nouveauté de l'idée consiste à utiliser un moteur Ringbom-Stirling qui utiliserait l'air comme élément de puissance et d'eau comme lubrifiant,

avec une puissance de pouvoir à son maximum par l'injection de vapeur à haute pression. De tels moteurs pourraient être fabriqués en convertissant les moteurs actuels de locomotives Stirling utilisant les mêmes pièces et les mêmes composants. Ainsi, il résulterait de très fortes économies en transformant graduellement l'équipement de locomotives vers la combustion en charbon plutôt que de le remplacer totalement par des locomotives à vapeur. Le rapport recommande la poursuite de l'étude du moteur Ringbom-Stirling et la mise au point, à titre démonstratif, d'un moteur de 20HP obtenu par la conversion d'un moteur diesel courant à deux cylindres. Le coût prévu est de 750 000$ répartis également sur une période de 3 ans.

La partie 11 du rapport fait une appréciation succinte du moteur à cycle de Stirling utilisant un combustible gazeux et appliqué à la propulsion de locomotives et, aborde divers aspects connexes de la question: manutention, stockage et danger du combustible liquéfié.

INTRODUCTION

The high price and limited availability of oil fuel is forcing a return to coal as the primary fuel. As this occurs the opportunities for application of Stirling engines are unlimited. Just as the last century belonged to the steam engine, and the present one to the internal combustion engine, so the next will belong to the Stirling engine.

Much of the coal will be used, along with nuclear energy, in baseload power stations to produce electric power. Cars, trucks and buses will use that power for propulsion with overnight charging by resistance heating of a lithium fluoride thermal battery to energize the Stirling automotive engine. Following its introduction by Meijer (1) subsequent studies (most recently Morgan (2)) have confirmed the feasibility and attraction of this concept as an alternative to electric storage batteries.

Other Stirling engines will use coal directly for stationary power generation, for heavy automotive applications in locomotives, surface mining equipment, off-highway vehicles, large, long-distance highway trucks and for marine use. There are no limits to maximum and minimum power capacities achievable with coal-fired Stirling engines. Given the will, virtually every oil-burning diesel engine is fair game for replacement by a coal-burning Stirling engine.

Limitations of space preclude the comprehensive discussion given elsewhere (Walker (3)) and we shall confine our subsequent discussion to large coal-burning Stirling engines (arbitrarily defined as greater than 1 MW output) applied to heavy automotive use for railroad locomotives and the larger off-highway vehicles.

LARGE STIRLING ENGINES

At the start it should be made clear there is no established experience of large Stirling engines. Most work to date has focussed on smaller engines up to 150 kW capacity. Walker (4) has briefly described the four-cylinder engine of 265 kW (360 horsepower) made by Philips for General Motors and the U.S. Navy in the early 1960's. Later the Electromotive Division of General Motors made a 300 kW (400 horsepower) 4-cylinder in-line, engine representing one bank of a Vee-8 engine of 800 horsepower, and did the design work for other larger Stirling engines. Hearsay has it that the German licensees MAN/MWM of the Philips Company have developed Stirling engines in the 600 kW (800 horsepower range) but no details of this work are known. Ishizaki (5) has briefly described a Japanese 750 kW (1000 horsepower) prototype engine.

Meijer (1) presented the results of a study made at the Philips laboratories for a 680 kW (900 horsepower) four-cylinder rhombic drive engine for stationary power having a capacity of 170 kW (225 horsepower) per cylinder, using helium as the working fluid.

Later Hoagland *et al* (6) extrapolated the established experience of the Philips licensee United Stirling with the well-developed P75 Siemens-Stirling engine. An outline design and the technology evaluated for a 1490 kW (2000 horsepower) Siemens-Stirling engine having 16 cylinders, four modules of the four-cylinder Siemens-Stirling 'square-four' arrangement shown in Figure 1. These encouraging results prompted the U.S. Department of Energy to sponsor further engineering studies of large coal-fired stationary power Stirling engines (AMT (7), General Electric (8)) in a programme reviewed by Holtz *et al* (9).

More recently, Walker (10) used the results of all these studies

to prepare Table 1 containing comparative data for 2200 kW (3000 horsepower) diesel and Stirling railroad locomotive engines.

Data for the ALCO Series 251 diesel was taken from the manufacturers (Bombardier, MLW) literature. Data for the Stirling engines were taken from the studies referenced above and adjusted for common heater and cooler temperatures of 800° and 70°C, respectively, output of 2200 kW (3000 horsepower) and a speed of 16.6 hz (1000 revs. per min.) for direct comparability with the diesel unit. The techniques for scaling and adjusting are fully described by Walker (10). Data for the rhombic engine was developed simply by adding more cylinders to the four-cylinder, 900 horsepower, engine described by Meijer (1). The result was a monstrous distortion to long-thin snake of an engine nearly twice the length of the 16-cylinder Vee diesel. In practice a 3000 horsepower rhombic engine would likely consist of four or six large cylinders. The important point to note is that even with such gross distortion the installation volume per unit power output is *exactly the same* for the diesel and rhombic Stirling engine.

The Siemens-Stirling engines derived from the Hoagland and General Electric studies both had 16 cylinders arranged in four modules of square-four cylinders. Hoagland followed the familiar United Stirling parallel-cylinder, double crank arrangement and General Electric favoured a Vee arrangement.

COAL COMBUSTION AND HEAT TRANSPORT SYSTEM

Fluidized bed coal combustors are preferred for large Stirling applications. They operate at relatively low temperatures so that nitrous oxide production is negligible. Limestone added to the bed prevents the emission of sulphur dioxide. Coal can therefore be readily burned with little or no pretreatment apart from cleaning, crushing and

screening to a uniform manageable size. Very high and predictable rates of heat transfer can be gained with fluidized bed combustors. Dunn *et al* (11) indicates heat transfer rates of 100 kW/m^2 of heat transfer surface for the boiling section of heat pipes immersed in the fluidized bed. Dunn *et al* (11) further suggests the maximum power density of fluidized bed combustors to be 650 kW/m^3. In round numbers, this corresponds to 4 m^3 per 750 kW (1000 horsepower) of engine output assuming 80 percent of the heat released in the bed reaches the engine, the engine conversion efficiency is 40 percent, and allowing 10 percent of the bed volume for heater tubes or heat pipe boiling sections.

It is possible for the engine heater tubes to be immersed directly in the bed, but the use of a sodium heat pipe is preferred to transport heat from the bed to the engine. Sodium boils in the bed section and the vapour condenses on the engine heater tubes. Very high rates of heat transfer in the boiling and condensing processes and heat fluxes as high as 15 kW/cm^2 can be achieved with little temperature difference between the combustor and the engine.

Use of the heat pipe heat transport system permits the fluidized bed to be relatively remote from the engine. Furthermore, there are no hot spots on the engine heater tubes on which the sodium vapour is condensing so the *mean* temperature of the tubes is also the *maximum* temperature, a 50 to 100°C increase in the mean temperature compared with directly heated tubes with the inevitable 'hot spots'. Moreover, the very high rates of heat transfer permit the tubes to be optimized from considerations of the engine internal geometry and to be smaller and shorter than those required for direct heating. The increase in heater temperature and decrease in internal dead volume increases the efficiency and power output of

the engine to a significant degree.

COST

Reliable data on the cost of Stirling engines is simply non-available. Hoagland *et al* (6) suggest that stationary Stirling engines burning distillate fuels would be 25 to 30 percent more than diesel engines of the same size, and, further, that Stirling engines using solid fuels would likely be 30 to 50 percent more expensive than on distillate fuels. The limiting range of costs is then 1.6 to 2.25 the cost of diesels. General Electric estimated costs of $230 to $297 per horsepower for a 1000 horsepower engine at a production rate of 1000 engines per year and compared this with the cost of $117 per horsepower for a 2000 horsepower diesel engine in current production.

IMPEDIMENTS TO APPLICATION

A coal-burning Stirling engine costing two or three times the oil-burning diesel it replaces can be readily accepted with a large difference in the cost of oil and coal, about five times per unit of heat at 1981 world prices. Furthermore, Table 1 shows the Stirling engine could likely fit the same installation envelope and have the same power and efficiency as the diesel it replaces. It would run without noise, has excellent part-load performance and favourable torque/speed characteristics.

Yet despite all this, the prospect of Stirling engines of the type discussed above supplanting diesels is not very bright. The reason, in a word, is seals, the Achilles heel of all high performance Stirling engines.

Seals are used on the piston rods to prevent leakage of the high pressure working fluid from the working space. For high performance, high power density engines, helium or hydrogen at high pressures must be

used. These gases are extremely difficult to contain and so replenishment must be carried out at intervals. The seals also prevent ingress of lubricating oil to the working space. This is even more important than preventing gas egress for the oil accumulates in the fine interstices of the regenerator, carbonizes and blocks the regenerator with consequent loss of power, overheating and burnout of the heater tubes.

Other seals on the pistons are used to separate the different working spaces of the Siemens-Stirling arrangement operating at cyclically different pressures. It is difficult to prevent some preferential 'blow-by' and as a consequence one space becomes starved of working fluid while its neighbour is gorged to excess.

Over the past 40 years a king's ransom has been invested in research and development on seals yet in 1978 Hoagland *et al (6) wrote '... Using state of the art seals United Stirling obtains leakage of hydrogen from the 4-piston rod seals of a 100 brake horsepower double-acting engine ranging from 0.1 to 0.2 grams per hour at full load. This life of their piston rings as well as the rod seals is currently about 4000 hours ...'.*

In North America the anticipated, and routinely achieved, life of a railroad diesel electric locomotive before major maintenance is 500,000 miles. The prospect of regularly replenishing helium working fluid (assuming it would be preferred to hydrogen for reasons of safety despite some loss of power) and changing seals every 4000 or even 10,000 hours is simply unacceptable.

In this desperate situation new approaches are clearly required and we present these below.

NOVEL APPROACHES: THE CONVERSION OF DIESELS TO STIRLING ENGINES

We have seen how large coal-burning Stirling engines could be designed and constructed to replace the oil-fuelled diesel engines in

railroad locomotives and so permit the gradual conversion and continued usage of existing locomotive fleets. The greatest impediment to implementation is the limited life of seals on the piston and piston rod and the need to replenish unfamiliar working fluids such as helium.

We propose to overcome these problems by the approaches outlined below and, to go further, to show how the existing railroad locomotive diesel engines may be converted to Stirling engines of a special type called Ringbom-Stirling engines.

Use Single-Acting Engines

Abandon the use of double-acting engines and revert to ensembles of single-acting engines on a common crankcase used in the Philips prototype engines of the 1960's.

Use Air as the Working Fluid

Use air as the working fluid instead of helium or hydrogen to eliminate the need for a hermetic gas seal. Reasonable leakage from the working space can be replenished from a reservoir fed by an engine driven air compressor.

Figure 2 shows the maximum possible thermal efficiency as a function of power density (brake horsepower per litre of engine displacement) for Stirling engines of 225 brake horsepower per cylinder using air, helium and hydrogen. This data was presented by Meijer (1) and resulted from the study of the 900 horsepower stationary power generator discussed above. The important point to note is that the endpoint of the curve for air indicates a power density of only 17 brake horsepower per litre of engine displacement. Yet this is precisely the same power density of 17.4 brake horsepower per litre engine displacement of the ALCO Series 251 diesel engine given in Table 1. Furthermore, the Stirling engine has a

power stroke every revolution whereas the diesel is a four-stroke engine and has a power stroke every two revolutions. The speed of Stirling engines at 400 revolutions per minute is then not too dissimilar to the 525 working strokes per minute of the diesel engine. However, to provide a direct replacement of the same speed, the Stirling engine could be accelerated to 1000 revolutions per minute without significant change in power and efficiency by relaxing the pressure to compensate for the increase in speed. Air was always used as the working fluid in regenerative engines until the early 1950's when Philips perceived the need to develop very high output engines comparable to small automotive gasoline and diesel engines.

The lower efficiency (30 percent) at maximum power of the Stirling engine is not significant for a locomotive engine spends little time at maximum power. Ephraim *et al* (12) published the typical duty cycle for a North American railroad locomotive which calls for the engine to be operating above 50 percent maximum power output for only 40 percent of the time. The loss in efficiency at high power is therefore compensated by the increased efficiency of the Stirling engine at lower power. The diesel engine reaches its maximum efficiency close to the rated output and the efficiency declines at lower output.

Use Steam Injection for Maximum Power

Use steam injection in the hot expansion space to obtain a power boost during periods of high power operation. This corresponds to the process of water or methanol injection used in Rolls-Royce Merlin engines for Spitfires in World War II when taking off or in combat.

In the steam injection Stirling engine superheated steam at high

pressure would be injected to the expansion space from a monotube, once-through, super-critical boiler immersed in the fluidized bed or sodium vapour heat pipe near the expansion space. The steam would be admitted through a valve operated automatically by the control system when the displacer was approaching top dead centre. The steam injected would condense in the regenerator and accumulate as liquid in the cold compression space. It would act as the lubricant for the piston rings and be pumped out of the working space by the piston seals. The water naturally descends from the hot expansion space to the cold compression space according to the Law of Minimum Energy States. Its presence in the compression space is beneficial for it acts as the seal lubricant. Furthermore, it improves the efficiency of the seals since they are now hydraulic rather than pneumatic. Another advantageous feature is that the water improves the heat transfer process in the compression space so that compression is closer to the ideal isothermal process.

Preliminary studies at the University of Calgary of two-phase, two-component working fluids have shown that very substantial increases in output may be gained compared with a gaseous working fluid. Figure 4 shows the theoretical work diagrams obtained by Walker *et al* (13) for two-phase two-component and gaseous working fluids. Injection of significant amounts of steam during periods of high power demand could likely double the power output of the engine but with substantial loss of efficiency.

Use Water as the Lubricant

Prevention of oil contamination of the working space is difficult to achieve in conventional Stirling engines. The problem can be eliminated by substituting water instead of oil as the lubricant. This also eliminates

the problem of separating oil from the 'return' water to be reused for steam injection Stirling engines.

Water is not as good as oil for lubrication but is tolerable in special circumstances. Its use in oxygen compressors and steel rolling mills has been routine for many years. Furthermore, the lubrication requirements in Stirling engines are very much less demanding than in diesel engines. This is particularly true for the piston rings which have a less arduous duty in Stirling engines. Moreover, the extremely high rates of pressure change experienced during combustion in diesel engines giving impact-like loading to the crankshaft and connecting rod bearings do not exist in Stirling engines. The pressure-time curves are virtually sinusoidal and the maximum rates of pressure change are very low in comparison with diesel engines.

Convert Diesel Engines to Stirling Engines

Figure 5 shows a Ringbom-Stirling engine. The piston is coupled to a crank and connecting rod and a free-displacer moves in an adjacent cylinder under the action of fluidic and inertia forces acting upon it. This type of engine was invented in 1905 by Ossian Ringbom, a Russian living in Finland. A complete discussion of the engine is given in the book by Walker and West (14). This arrangement permits existing internal combustion engines to be converted to Ringbom-Stirling engines. Figure 6 shows an early Ringbom-Stirling engine made at the University of Calgary (Walker (15)) by converting a Honda industrial engine. More recently, a prototype 1 kW Ringbom-Stirling engine has been made by Sunpower Inc. (Beale (16)) pioneers of free-piston Stirling engines.

Applying the principle of conversion the ALCO Series 251

diesel engine shown in Figure 7 might appear as the Ringbom-Stirling engine shown in Figure 8. A more comprehensive discussion of various aspects of this conversion is given by Walker (3) and (10).

THE COAL-BURNING STIRLING LOCOMOTIVE

Application of the Ringbom-Stirling engine described above to a railroad locomotive is illustrated in Figure 9. The locomotive is modelled after the Bombardier MLW M630 six-axle 3000 horsepower diesel electric locomotive in current usage.

Twin-fluidized bed coal combustors are located beneath the frame of the locomotive between the bogies. There is space also for a small buffer coal store but for long hauls the locomotive would tow a coal tender in the same way diesel locomotives tow fuel tankers.

Heat from the fluidized beds are conveyed to the engine by sodium heat pipes clustered in well-insulated conduits. The additional cooling required for Stirling engines is obtained by enlarging and double-banking the existing cooling water 'radiators'.

An electrically driven air compressor and reservoir provides compressed air for initially fluidizing and heating the combustors, for starting and for charging or recharging the Stirling engine. A large engine driven fan supplies air to the fluidized bed when the system is operating. A large recuperative exhaust gas/inlet air preheater replaces the customary diesel engine muffler.

Apart from the above changes, replacements or additions, the rest of the Stirling engine locomotive is exactly the same as the present diesel electric locomotive. The principal difference to the environment of the conversion to coal-burning Stirling locomotives is the absence of noise and a very much reduced level of air pollution.

Kneiling (17) suggests that conventional diesel electric units on long runs with an oil tank car attached consume 2 (U.S.) gallons of diesel oil per 1000 gross ton miles. Two power units hauling a 5000 ton train uses 10 (U.S.) gallons per loaded ton mile. A 1000 mile haul and empty return trip consumes 15000 (U.S.) gallons. The same road trip with two coal-burning Ringbom Stirling engines would consume 75-80 tons of coal (Walker (10)).

CONCLUSION

Given the substantial price differential between coal and oil, the future for coal-burning Stirling engines in heavy traction applications appears brighter than ever before. It appears feasible to convert existing diesel-electric railroad locomotives to coal-burning Stirling-electric locomotives. Moreover, it appears possible to modify the diesel engines to Ringbom-Stirling engines. Given the large inventory of existing diesel-electric locomotives the potential savings of conversion compared with outright replacement are enormous.

ACKNOWLEDGEMENTS

The study of Stirling engines applied to railroad locomotives described herein was supported by Transport Canada Transportation Development Centre, Montreal. Technical information was received from Bombardier MLW, Montreal.

LIST OF REFERENCES

1. Meijer, R. Prospects for the Stirling Engine for Vehicular Propulsion. Philips Tech. Rev., Vol. 31, No. 5/6, pp. 168-185, 1970.
2. Morgan, D.T. Thermal Energy Storage for the Stirling Powered Automobile. Dept. of Energy Report, No. ANL-K-78-4135, 1978, (available N.T.I.S., Springfield, Va.).
3. Walker, G. Large Stirling Engines. (monograph in preparation), University of Calgary, Alberta, Canada.
4. Walker, G. Stirling Engines. Oxford University Press, 1980.
5. Ishizaki, Y. Stirling Engine Development in Japan. Paper No. 799437, Proc. 14th Inter. Soc. Energy Conv. Eng. Conf., pp. 2035-38, Boston, Aug., 1979.
6. Hoagland, L.C. and Percival, W.H. A Technology Evaluation of the Stirling Engine for Stationary Power Generation in the 500 to 2000 Horsepower Range. Dept. of Energy, Report No. ORO/5392-01, 1978, (available N.T.I.S., Springfield, Va.).
7. Anon. (Advanced Mechanical Technology Inc.). Design and Development of Stirling Engines for Stationary Power Generation in the 500 to 3000 Horsepower Range. Dept. of Energy, Report No. DOE/ET/15207-T1, 1980, (available from N.T.I.S., Springfield, Va.).
8. Anon. (General Electric). Design and Development of Stirling Engines for Stationary Power Generation Applications in the 500 to 3000 Horsepower Range. Dept. of Energy, Report No. DOE/ET/15209-T1, 1980, (available from N.T.I.S., Springfield, Va.).
9. Holtz, R.E., Marciniak, T.J. and Uherka, K.L. DOE Stationary External Engine Programme: Status Report. Paper No. 799242, pp. 1120-1123, Proc. 14th Inter. Soc. Energy Conv. Eng. Conf., Boston, Aug. 1979.

10. Walker, G. The Stirling Engine for Railroad Locomotive Application. Report for Transport Canada, Transportation Development Centre, Montreal, 1981, Report No. TP2903E.

11. Dunn, P.D., Rice, G. and Thring, R.H. Hydraulic and Rotary Drive Stirling Engines with Fluidized Bed Combustion/Heat Pipe Systems. Paper No. 759141, pp. 942-947, Proc. 10th Inter. Soc. Energy Conv. Conf., Newark, Del., Aug. 1975.

12. Ephraim, M. and Addie, A.N. Diesel Traction in North America. Jnl. Railway Eng., Inst. of Mech. Eng., London, Vol. 14, pp. 73-84, 1975.

13. Walker, G. and Agbi, B. Thermodynamic Aspects of Stirling Engines with Two-Phase Two-Component Working Fluids. Trans. Can. Soc. Mech. Eng., Vol. 2, No. 1, pp. 1-8, 1974.

14. Walker, G. and West, C. Free-Piston and Liquid-Piston Stirling Engines. (monograph in press) (to be published 1982).

15. Walker, G. Stirling Cycle Machines. Oxford University Press, 1973.

16. Beale, W. Private Communication. Sunpower Inc., Athens, Ohio.

17. Kneiling, J.G. Trains for Tomorrow. Trains, July, pp. 28-33, August 1977, pp. 44-50.

Column No.	1	2	3	4	5	6	7
	Series M251 Diesel	Meijer Rhombic Drive Engine B-Fig. 11	Revised Meijer Rhombic	Hoagland *et al* (1978) Engine	Revised Hoagland Engine	General Electric (1980)	Revised General Electric
Horsepower (gross)	3050	900	3094	2000	3000	1000	3000
No Cylinders	16	4	14	16	16	8	16
HP/Cylinder	190.6	225	221	125	187.5	125	187.5
Speed RPM	1050	810	1000	850	1000	1200	1053
Cylinder Bore (ins)	9	-	-	7.18	9.55	7	7.97
(mm)	*228.6*	-	-	*182*	*242.6*	*177.8*	*202.4*
Piston Stroke (ins)	10.5	-	-	4.0	5.32	4.0	4.55
(mm)	*266.7*	-	-	*102*	*135.1*	*101.6*	*115.6*
Swept Vol/Cyl (in^3)	668	-	-	162	380	153.94	227
(litre)	*10.95*	-	-	*2.65*	*6.22*	*2.52*	*3.72*
Total Swept Vol (in^3)	10688	-	-	2592	6080	1231	3632
(litre)	*175.20*	-	-	*42.4*	*99.52*	*20.16*	*59.5*
Engine Weight Drag (lbs)	40800	-	-	32000	39336	12222	36600
(kg)	*18507*	-	-	*14500*	*17880*	*5555*	*16636*
Weight/HP (lbs/HP)	13.37	-	-	16	13.1	12.2	12.2
(kg/HP)	*6.06*	-	-	*7.25*	*7.25*	*5.55*	*5.55*
Length (ins)	191	90	317	143	164	72	166.5
(m)	*4.89*	*2.3*	*8.05*	*3.632*	*4.16*	*1.82*	*4.22*
Breadth (ins)	60.5	39.4	39.4	50	57.5	84	96.6
(m)	*1.55*	*1.0*	*1.0*	*1.270*	*1.45*	*2.13*	*2.45*
Depth (ins)	96.5	94.5	94.5	68	78.2	78	89.7
(m)	*2.47*	*2.4*	*2.4*	*1.727*	*1.98*	*1.37*	*2.27*
Installation Volume (ft^3)	645.3	193.9	683	281.4	421.8	273	834.9
(m^3)	*18.75*	*5.52*	*19.32*	*7.96*	*11.94*	*5.31*	*23.46*
Vol. per HP (ft^3/HP)	0.215	0.215	0.215	0.1406	0.1406	0.273	0.273
(m^3/HP)	*.0061*	*0.006*	*0.006*	*0.0039*	*0.0039*	*0.0053*	*0.00782*
HP/Unit Swept Volume (HP/in^3)	.285	.655	.655	.7716	.493	.813	-
(HP/litre)	*17.40*	*40*	*40*	*47.16*	*30.14*	*49.58*	*50.4*
Max. Thermal Efficiency (%)	37	39	39	39.6	35	40	40
Heater Temperature (°C)	-	700	800	800	800	760	800
Cooling Water (°C)	-	25	70	30	70	80	70
Working Gas	Air/Fuel	Helium	Helium	Helium	Helium	Helium	Helium
Mean Operating Pressure (psi)	-	1552	1261	2030	1297	1470	1470
(MPa)	-	*10.7*	*8.7*	*14.00*	*8.94*	*10.1*	*10.1*
Max. Pressure (psi)	1450	-	-	-	1.733	1984	1984
(MPa)	*1.0*	-	-	-	*11.95*	*13.68*	*13.68*
Min. Pressure (psi)	-	-	-	-	866	955	955
(MPa)	-	-	-	-	*5.97*	*6.58*	*6.58*
Mean Eff. Pressure (psi)	215	-	-	-	-	-	-
(MPa)	*1.48*	-	-	-	-	-	-
Engine Type	Diesel	Stirl. Rhomb.	Stirl. Rhomb.	Stirl. Siem.	Stirl. Siem.	Stirl. Siem.	Stirl. Siem.
Column No.	1	2	3	4	5	6	7

TABLE 1

Comparative Data for 3000 Horsepower Diesel and Stirling Locomotive Engines

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Figure 9 The Coal-Burning Ringbom-Stirling Powered Railroad Locomotive.

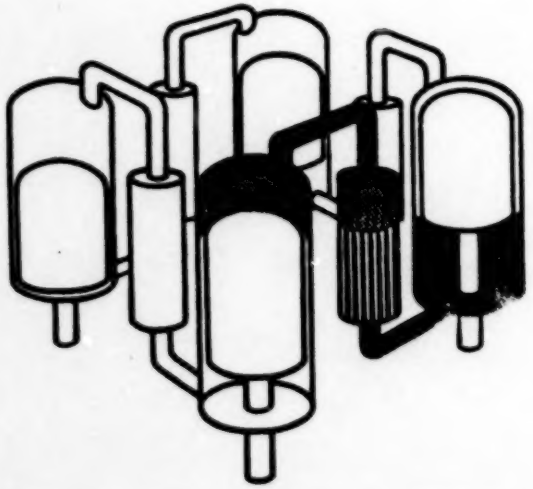


Figure 1: The Square-Four Arrangement of the Siemens-Stirling Engine.

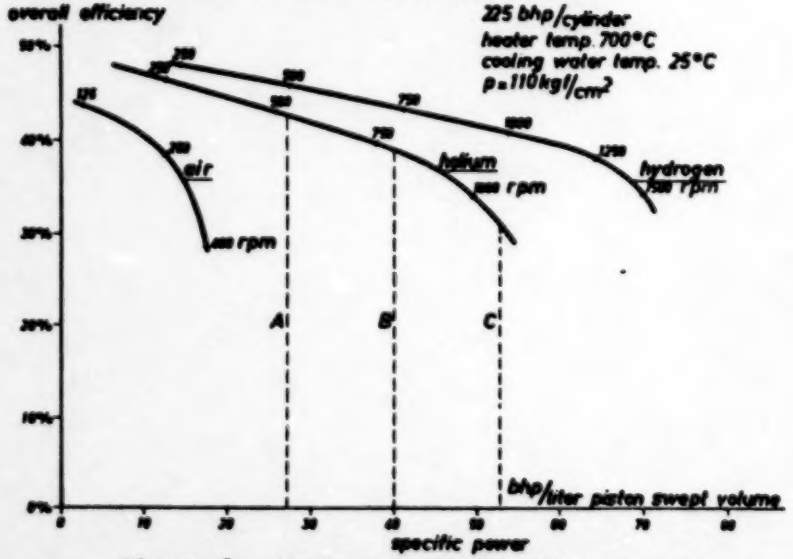


Figure 2: Maximum Possible Efficiency of a Stirling Engine as a Function of the Power Density with Air, Helium and Hydrogen Working Fluids, an Output of 225 Brake Horsepower per Cylinder and a Mean Working Fluid Pressure of 10.7 MPa (110 kg_f/cm^2).

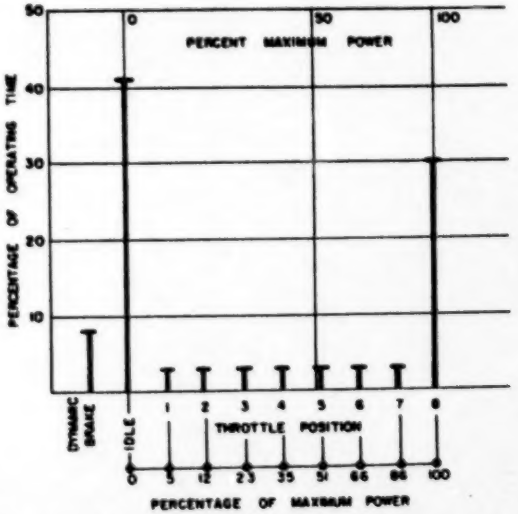
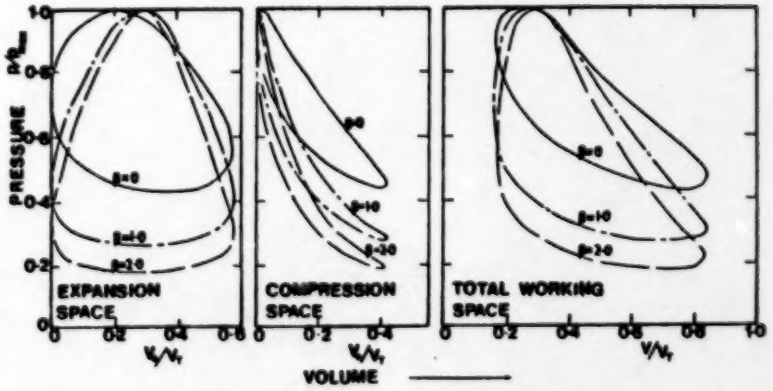


Figure 3: Representative Duty Cycle for North American Railroad Locomotive, (after Ephraim *et al* (12)).



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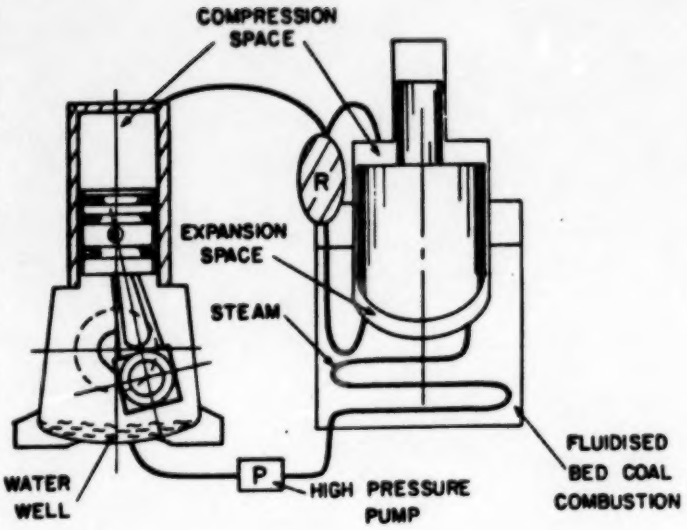

Figure 5: The Ringbom-Stirling Engine.

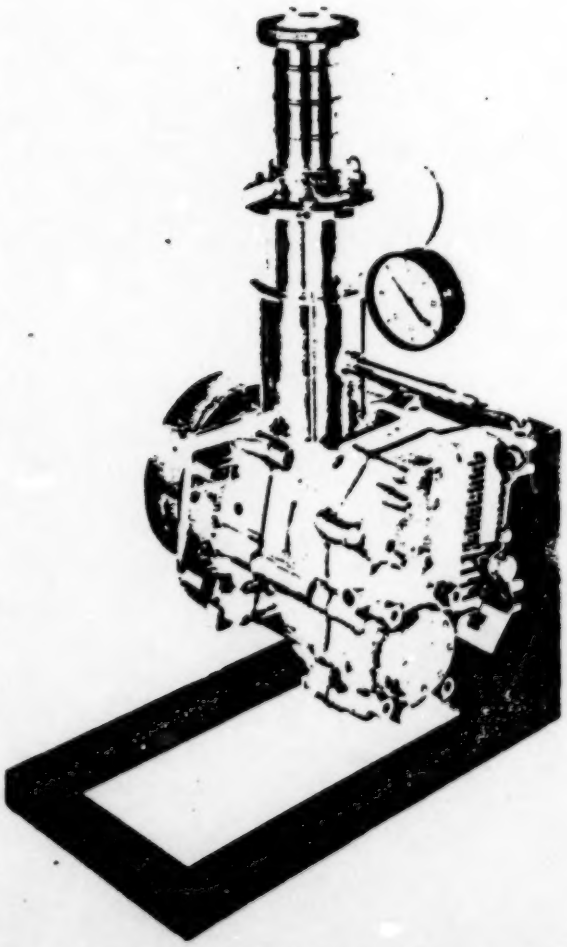

Figure 6: Early Ringbom-Stirling Engine Made by Converting a Honda Industrial Engine.

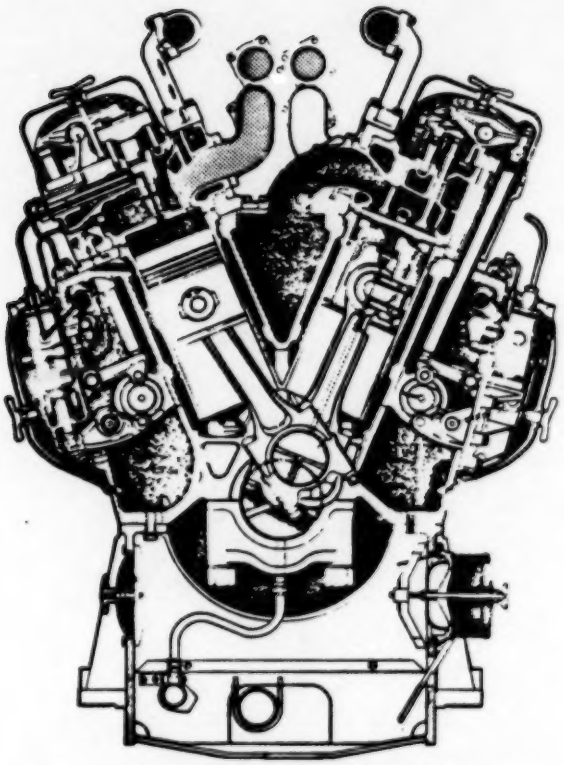

Figure 7: Cross-Section of MLW Series 251 Four-Stroke Diesel Vee Engine, (after Bombardier MLW).

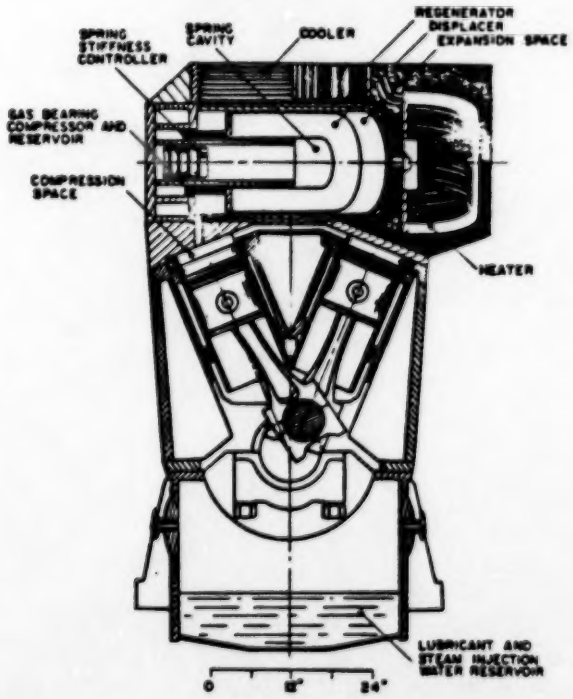

Figure 8: Ringbom-Stirling Engine Based on Conversion of the MLW Series 251 Diesel Engine with One Displacer Cylinder per Pair of Engine Cylinders.

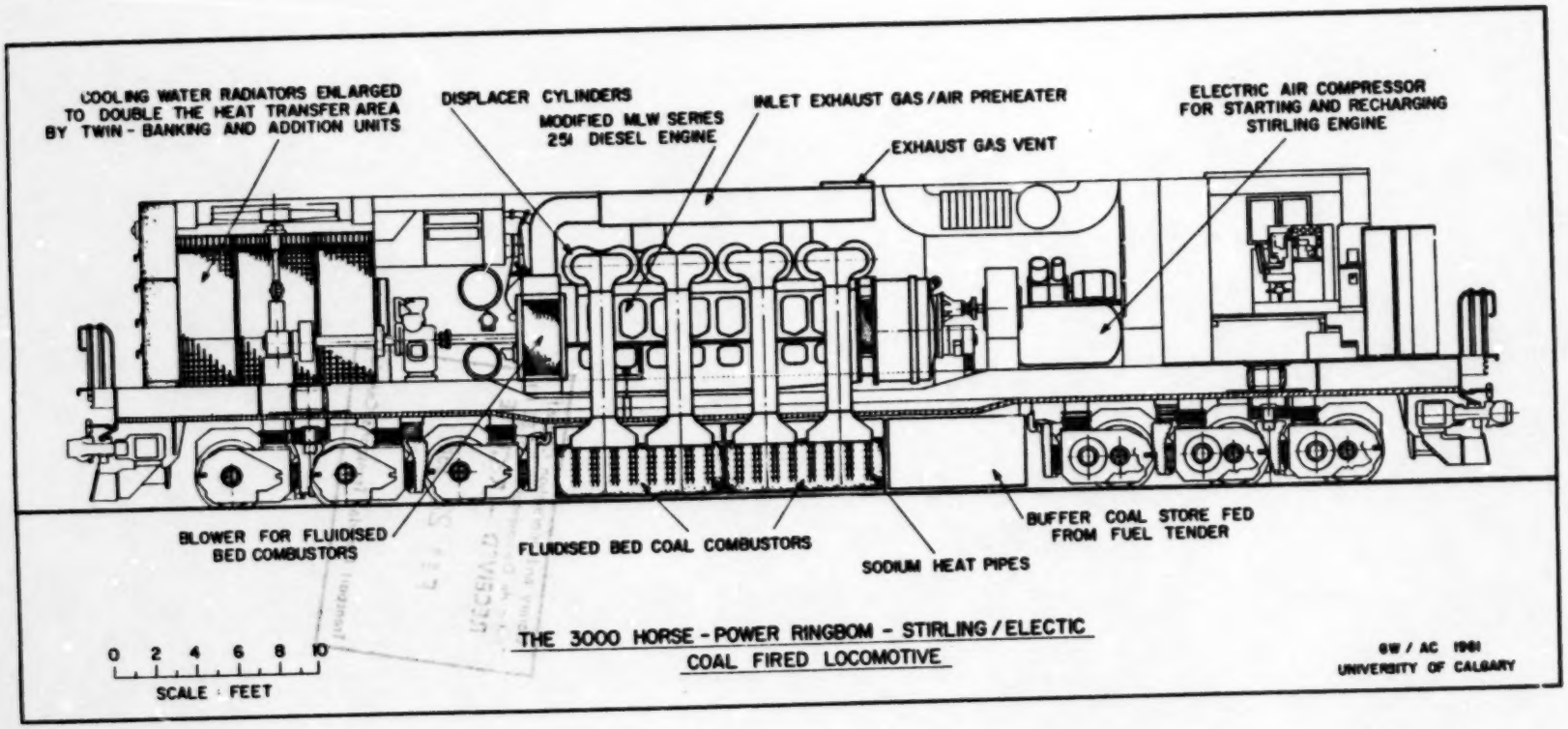
COOLING WATER RADIATORS ENLARGED TO DOUBLE THE HEAT TRANSFER AREA BY TWIN - BANKING AND ADDITION UNITS
DISPLACER CYLINDERS
MODIFIED MLW SERIES 251 DIESEL ENGINE
INLET EXHAUST GAS / AIR PREHEATER
EXHAUST GAS VENT
ELECTRIC AIR COMPRESSOR FOR STARTING AND RECHARGING STIRLING ENGINE
BLOWER FOR FLUIDISED BED COMBUSTORS
FLUIDISED BED COAL COMBUSTORS
SODIUM HEAT PIPES
BUFFER COAL STORE FED FROM FUEL TENDER
0 2 4 6 8 10
SCALE : FEET
THE 3000 HORSE - POWER RINGBOM - STIRLING / ELECTIC
COAL FIRED LOCOMOTIVE
GW / AC 1981
UNIVERSITY OF CALGARY

Figure 9